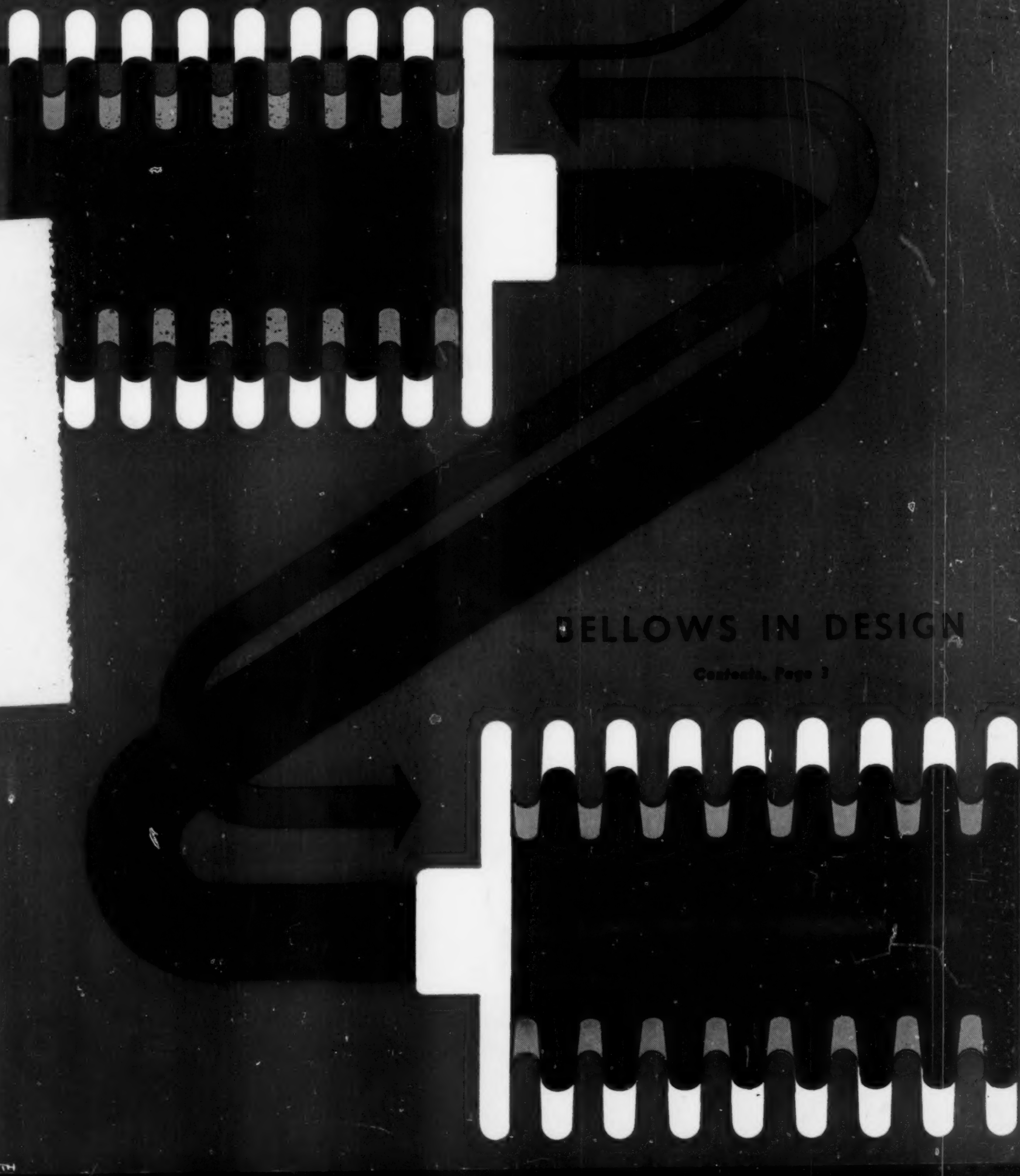
MACHINE DESIGN

January 1954



BELLOWS IN DESIGN

Contents, Page 3

THE PROFESSIONAL JOURNAL
FOR ENGINEERS AND DESIGNERS

MACHINE DESIGN

January, 1954
Vol. 26—No. 1

Over the Board

Our Silver Anniversary

The sharper-eyed among you may have noticed that this is the first issue in Volume 26 and concluded that we have therefore completed a quarter-century of publication. However, because Volume 1 contained only five issues (beginning September 1929) we have to wait until the August 1954 issue is out before we can look back to 300 issues—a full 25 years of monthly issues. We confess to a feeling that birthdays of themselves mean nothing if regarded solely as a measure of survival. What the celebrant has done with the time is much more significant, and on that score we feel entitled to hold our heads high in this, our silver anniversary year.

A Publishing Revolution

Speaking of birthdays, our "median reader" is 43 years old, which means that half of our present readers were under 18 when MACHINE DESIGN started—too young to be readers or to appreciate the publishing revolution which the infant "professional journal for engineers and designers" initiated. Since the birth of MACHINE DESIGN other publications directed to design engineers have appeared and existing publications have changed their slant to capitalize on the growing influence of design engineers in important decisions. It's all good healthy competition which we welcome for its stimulating effects.

New Naval Strategy

Did you see the picture on Page 184 of our November issue? It showed a "giant nutcracker . . . capable of crushing the bows of a destroyer like a nutshell . . . built in Britain for the Royal Navy." Benjamin Ellison of the W. L. Maxson Corp. feels that the item leaves an important tactical problem unanswered and writes: "How does the Royal Navy lure enemy destroyers into its giant nutcracker? Various members of our force who are adept at naval strategy and tactics have proposed the following solutions: (a) Paint the device to resemble water; (b) bait the device with a chocolate bar; (c) camouflage this device to resemble the world's first male destroyer and, as all existing vessels are shes, the enemy will beat a path to its maw." Any other ideas?

This Month's Cover

Result of cross-breeding a helical spring with a hydraulic cylinder might be called a "sprinder" or a "cyling," but practical engineers are content to call such hybrids bellows. You'll see a pair of them in George Farnsworth's front cover design, employed simply as a motion-transmission mechanism. Many other uses of this versatile machine element are discussed in J. H. Howard's article beginning on Page 137, which is the most comprehensive guide to the selection and application of bellows we have ever seen.

MACHINE DESIGN

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Published on the seventh of each month. Subscription in the United States, possessions, and Canada: One year $10. Single copies, $1.00. Other countries one year, $20.

Acceptance under Section 34.64 P. L. and R. authorized.

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Advertising and editorial content itemized for convenience when studying specific design problems

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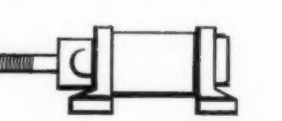
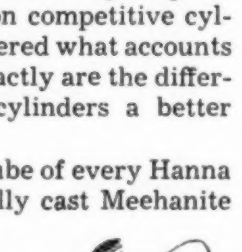
WHAT IS IT THAT MAKES ONE CYLINDER *BETTER* THAN ANOTHER?



When people set out to buy silverware, they accept without question the difference in price between sterling and plate. Without being silversmiths, they know that the price variance is justified by the intrinsic difference in quality, performance and life expectancy. This is equally true in the case of automobiles. Or in fine furniture. It's no less true in the case of cylinders.

There's a marked difference in cylinders, too. As in the case of many other products, it isn't always apparent to the naked eye. It's there, nonetheless, and over a period of time the difference makes itself known. If, on occasion, you have been quoted a higher price on Hanna cylinders than on competitive cylinders, you may have wondered what accounts for the difference. What exactly are the differences that make Hanna cylinders a better investment?

Well, for one thing, the tube of every Hanna H-P cylinder is centrifugally cast Meehanite (1-1/16" av. thickness), accurately bored and honed for low friction and perfect oil seal. Cast Meehanite is used because it lends cylinder tubes greater dimensional stability and more stubborn resistance to external damage.

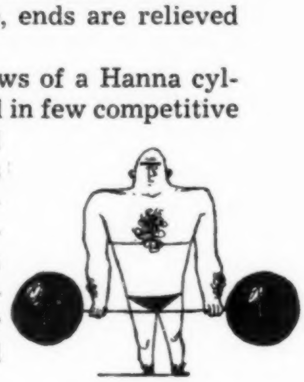


There's a pronounced difference, too, between the head of most cylinders and the Hanna cylinder head. In the Hanna product, large, metal-to-metal contact area between cylinder head and tube prevents cocking and assures squareness of mounting surfaces to the bore.

Hanna rod packings are designed for a more efficient seal and longer, trouble-free cylinder life. The spring-backed rod packings provide compensation for friction and wear.

The piston and rod of a Hanna cylinder are constructed of either a single piece of high carbon steel to assure concentricity and maximum strength, or assembled with shrink-fit welded construction and machined after assembly. In the latter case, ends are relieved to prevent wall scoring.

Even the assembly screws of a Hanna cylinder boast a feature found in few competitive cylinders. The heat-treated alloy socket head screws are closely spaced for maximum strength and countersunk into heads for a clean-cut appearance. Non-tie rod construction further enhances the streamlined design.



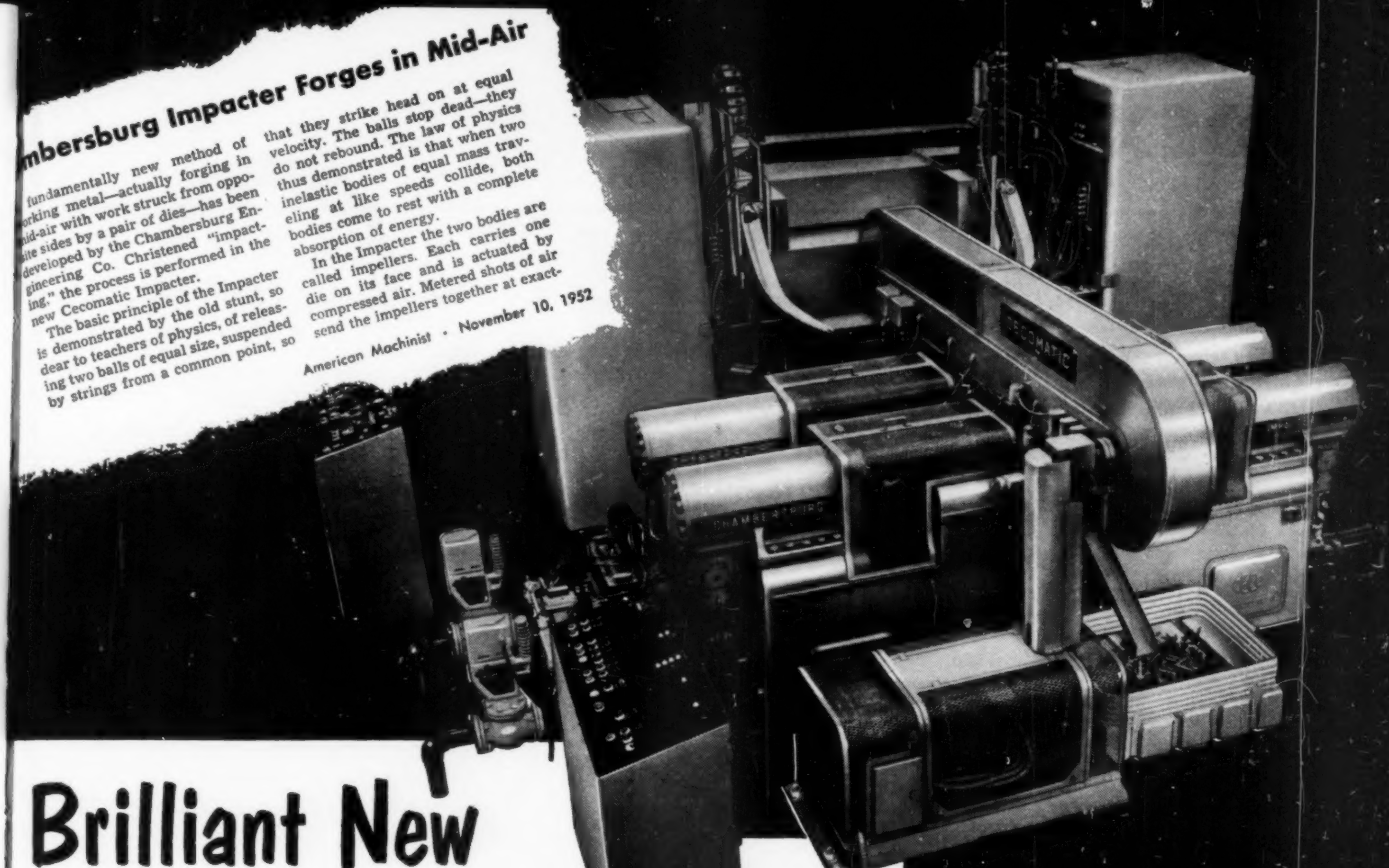
These, in the main, are the features that set Hanna cylinders apart from most other cylinders. Once you've used Hanna H-P cylinders, we're sure you'll agree their improved performance and longer life more than justify any small difference in cost. May we suggest that you bear this in mind next time you invest money in cylinders?

Send for your copy of the new Hanna H-P cylinder catalog

Hanna Engineering Works

1751 ELSTON AVENUE • CHICAGO 22, ILLINOIS

HYDRAULIC AND PNEUMATIC EQUIPMENT • CYLINDERS • VALVES • RIVETERS



...mbersburg Impacter Forges in Mid-Air

...fundamentally new method of ...orking metal—actually forging in ...id-air with work struck from oppo...site sides by a pair of dies—has been developed by the Chambersburg Engineering Co. Christened "impacting," the process is performed in the new Cecomatic Impacter.

The basic principle of the Impacter is demonstrated by the old stunt, so dear to teachers of physics, of releasing two balls of equal size, suspended by strings from a common point, so that they strike head on at equal velocity. The balls stop dead—they do not rebound. The law of physics thus demonstrated is that when two inelastic bodies of equal mass traveling at like speeds collide, both bodies come to rest with a complete absorption of energy.

In the Impacter the two bodies are called impellers. Each carries one die on its face and is actuated by compressed air. Metered shots of air send the impellers together at exact-

American Machinist · November 10, 1952

Brilliant New Machine Design

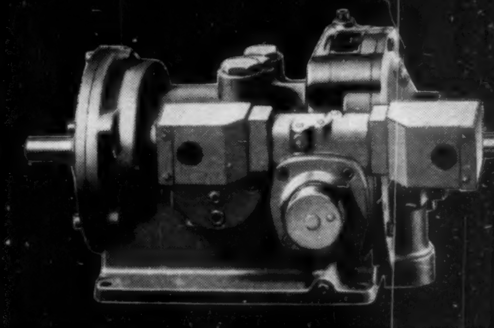
CRACKS "BARRIER" PROBLEM with OILGEAR FLUID POWER

Once the "forge-it-in-mid-air" idea occurred to the Chambersburg Engineering people, there were two "barrier" problems to be smashed before the idea became a brilliant reality. Something of the seriousness of the problems is indicated by the fact they spent 10 years to get the right, precise integration of movements and forces required.

One problem, that of getting the blanks to be forged into the right place at the right time for the mid-air forging blow, was eventually solved by recourse to *Any*-Speed Oilgear Fluid Power. The standard totally enclosed Oilgear Transmission with integral electro-hydraulic control gave the designers of the Cecomatic Impacter remote, interlocked, precise control of the conveyor . . . quick, cushioned acceleration . . . high traverse speed . . . fast, cushioned deceleration . . . smooth stop and dwell in forging position; all at the speed and accuracy they needed . . . just as so often in the past Oilgear has given *other* machine designers what *they* wanted.

Maybe your problem can be solved, your machine's performance improved by the smooth, swift acceleration and deceleration of the *Any*-Speed Oilgear. Maybe you need its extreme precision of controllability . . . or any of a dozen other remarkable characteristics. In addition you get simplicity, ease of assembly into your machine, ruggedness, dependability, accessibility, a unique freedom from maintenance.

But whatever your need, *you have not exhausted all possibilities in machine design if you have not investigated Oilgear Fluid Power Pumps, Motors and Transmissions exhaustively*. You too may have a world beater in your hands. THE OILGEAR COMPANY, 1568 W. Pierce St., Milwaukee 4, Wis.



Cecomatic Impacter doubles, triples, quadruples—octuples forging production! No jar or vibration. Less metal stress and die wear. Blanks load into conveyor automatically. And automatically the Oilgear type AXB-33 Variable Speed Transmission accelerates conveyor swiftly, smoothly, it smoothly stops with forging blank "dead on target," waits for the forging blow, then accelerates forged part away and new blank into position. Oilgear is adjustable up to 40 cycles per minute, is easily set to any required index distance which is then maintained under complete automatic control with unvarying precision.

OILGEAR

OILGEAR

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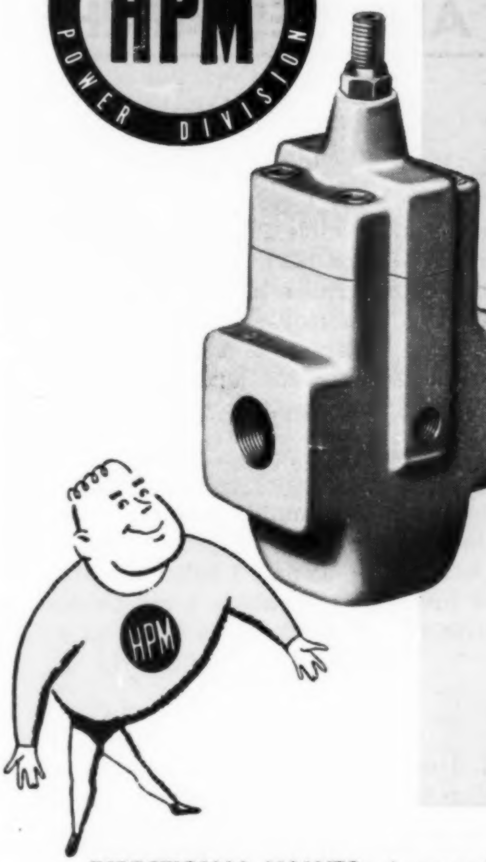
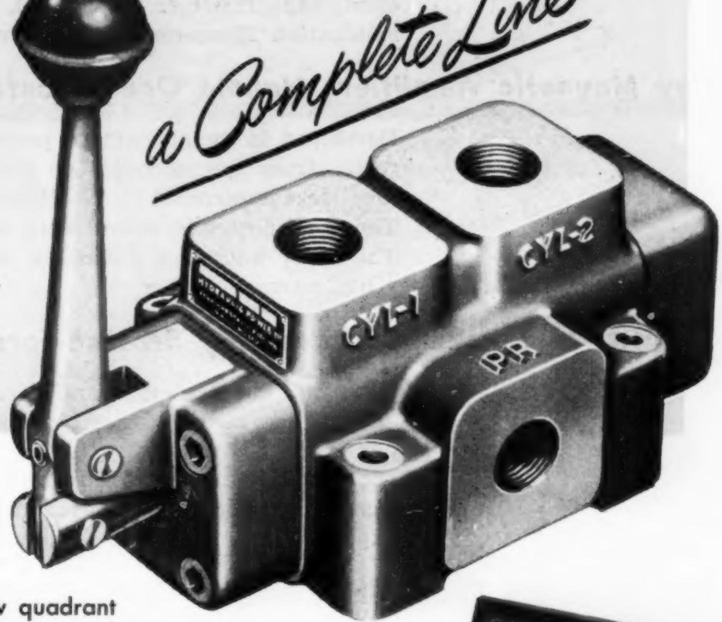
HPM
HYDRAULIC POWER DIVISION

VALVES

a Complete Line





DIRECTIONAL VALVES—4-way and 2-way threaded valves, ¼" to 1½" in size for 2000 P.S.I. service • Manual, cam, foot, solenoid, oil and air pilot operation • Six spool styles • Stem, knob, lever and a new quadrant type control • Same models, types of operation and sizes also available with flange connections for 3000 P.S.I. • An equally complete line of gasket mounted valves, ½" to 4", 2000 and 3000 P.S.I. • Also rotary pilot valves • 3175 valve models with modern, improved styling for a maximum of service.

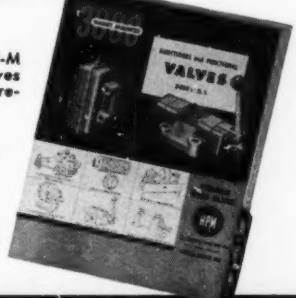
FUNCTIONAL VALVES—Relief, sequence, unloading, pressure reducing, counter-balance, flow control and check valves • Threaded connections, ¼" to 1½" sizes for 2000 P.S.I. • Flanged connections, ½" to 2½" sizes for 3000 P.S.I. • Gasket mounted, ½" to 2½" for 2000 and 3000 P.S.I. • Smooth, balanced control • Minimum pressure differential required for operation.

Write for the new H-P-M Catalogs covering Valves best suited to your requirements.

Cat. No. 801
2000 P.S.I. Threaded

Cat. No. 821
3000 P.S.I. Flange

Cat. No. 841
3000 P.S.I. Sub-Plate



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POWER
DIVISION
THE HYDRAULIC PRESS MFG. CO.
MOUNT GILEAD, OHIO, U. S. A.

1096 Marion Road

A Factor in the Field of Hydraulics Since 1877!

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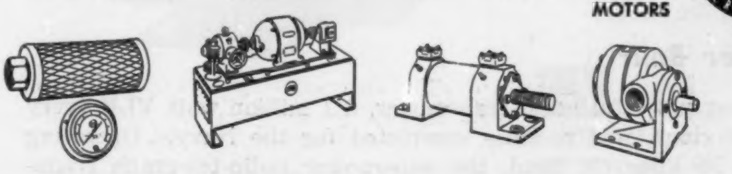


You know what you're buying when you specify HPM

MACHINE DESIGN Topics ... IN ENGINEERING AND RESEARCH

A Tinkertoy Project—for $2,175

A design kit containing all the equipment to assemble Project Tinkertoy units (see MACHINE DESIGN, November, 1953, Page 192) can now be purchased—for $2,175. Made available to help companies initiate design or redesign of electronic equipment for Project Tinkertoy automatic assembly, the kit includes all hand tools, furnaces, wafers, pins, etching equipment, tape resistors, capacitors, etc., to build over 200 modules. Communication Measurements Laboratory Inc. is the supplier.

Tiny Magnetic Amplifiers Handle One-Billionth Watt Signals

Developed in an attempt to provide electronic-free amplifiers working directly from thermocouple or similar low-energy device output, magnetic amplifiers announced by Westinghouse will operate on only 1 billionth watt. The amplifiers are wound with size 49 insulated wire about 1 mil thick. The core, wound of Permaloy strip 0.00012-inch thick, is as small as a pencil eraser.

Foil for Honeycomb Successfully Stretch-Formed

Type 302 stainless-steel and Inconel foil only 0.005-inch thick have been successfully stretch-formed, the first time that material of such thin gage has been so processed. Used for laminated honeycomb insulating panels, the material has previously been formed by regular punch-and-die operations. According to Longren Aircraft Co., the new process is much less expensive.

Are Selenium Rectifiers Toxic?

During the past year, several articles have appeared labeling selenium rectifiers—extensively used in radio and television sets—as toxic. A recent Navy Bureau of Ships *Journal* article points out the fallacies in such reports. Commenting on Navy policy, the article says, ". . . selenium rectifiers have been in general use for the past 15 years . . . evidence that little or no danger exists in their use. . . . Unless positive information develops showing the existence of danger, selenium rectifiers will continue to be used. . . ."

Silicone Rubber Stays Flexible at —120 F

Remaining flexible at temperatures down to −120 F, a silicone rubber compound developed by the Chemical Div. of GE is said to achieve this low-temperature flexibility without sacrifice of other desirable properties. Heat resistance and insulation ability, for example, are said to be unimpaired.

Most Powerful Radio Transmitter Built

The world's most powerful radio transmitter, a 1 million watt VLF (very low frequency) giant, was recently completed for the Navy. Operating in the 14.5 to 35 kilocycle band, the superpower radio-telegraph transmitter developed by Radio Corp. of America will be able to flash radio signals to any quarter of the globe. Signals will even penetrate to submarines cruising below surface, and can be heard in polar regions despite

frequent magnetic storms and ionosphere disturbances. Four powerful electron tubes develop the output, compared with eight tubes required in previous equipment to develop half the power. Antennas consist of ten catenary spans of cable ranging from 5640 to 8800 feet in length supported from twelve 200-foot towers. Towers are located on two mountains north and south of the 725-acre transmitter area near Arlington, Wash.

Bean-Sized Germanium Photocell Is Exceptionally Sensitive

Still in the development stage, a miniature electric eye announced by General Electric is slightly thicker than a pencil lead and only 3/8-inch long. More sensitive to light than vacuum photocells a hundred times larger, the germanium photocell has a comparatively large output; relays can be operated directly. The cell, a p-n junction type, consists of a metal cartridge housing a glass lens and a germanium wafer, in contact with a metallic button of indium.

Protects Tires Against Smog and Smoke

According to Firestone Tire & Rubber Co., a new material is now being used in tires to protect them against the deteriorating effects of smog, smoke, and chemical fumes. The new material, mixed into the rubber at the time of making the tire, is said to provide a film-like coating over the surface of the tire, and provides extra protection from weather checking and cracking.

Motors and Generators Built with Iron in Core Only

For one military application, a sizable number of ac and dc motors and generators are being built almost entirely without iron. Frames are made of aluminum fabricated by shielded-arc welding, according to Westinghouse, and shafts are made of stainless steel. Only the cores are iron, in most cases producing machines which are as small and somewhat lighter than their conventional counterparts.

Xeroradiography—Newest Industrial X-Ray Technique

Use of radiography for inspection of castings may get another boost by the development of Xeroradiography, a new process being tested at Alcoa's Cleveland Research Div. Developed jointly by G-E's X-Ray Dept. and the Haloid Co., the process is a further development of Xerography, and employs a reusable, specially coated aluminum plate and dry powder which is attracted by a static electricity charge to produce the image. Chief advantage of the process is its speed—the X-ray image is available for viewing within 45 seconds as compared with 1 hour required to develop and dry conventional X-ray film.

Color Changes in Paints Reveal Temperature

Over a number of years, research workers at the Naval Research Laboratory have developed a series of compounds that will give a color indication of temperature. Temperature-sensitive compounds have been developed that will record temperatures from 50 to 278 C (112 to 532 F) with some gaps, and work is currently under way to extend this range in the lower and medium temperatures. Some color-changing paints and crayons have recently been made available as imports from Germany.

MACHINE DESIGN
JANUARY 1954

Dungeons of Industrial Oblivion?

DO DESIGN engineers lead drab lives? We recently read in *The Houghton Line* this explanation of the alleged shortage of machine designers: Very few young men would consider long hours of study leading only to "an uneventful existence in the dungeons of industrial oblivion."

Perhaps the picture is overdrawn. It ignores the thrills that the design engineer experiences in successfully meeting the far from uneventful challenges of difficult engineering problems, the excitement of the initial test run, the personal pride and satisfaction of seeing the child of his brain take shape and become an effective contribution to man's progress and happiness.

But the author of the quoted statement, Ray Wilburth, makes this highly pertinent point: When a doctor, lawyer, writer, even an architect or civil engineer, does a significant job, he and his work are well headlined; but industrial achievement, if covered at all, is relegated to a brief item on the financial page with never a word about the engineers responsible for design.

We don't agree with those who would have engineers try to emulate the medical and legal practitioners, some of whose activities have hardly enhanced the reputation of their professions. But the impact of the design engineers' work on everyday living, considering its importance, is impersonal to an unnecessary degree. While design engineers may often seem to be modest, self-effacing people, we suspect that they are no different from other human beings in craving appreciation and the public recognition that is their due.

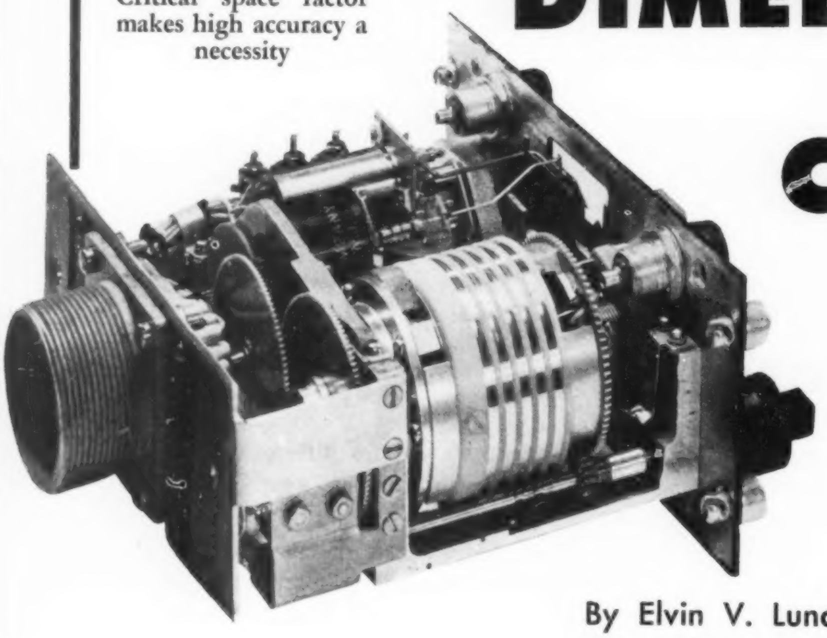
The key lies in the hands of top management in companies employing engineers. A comprehensive public relations program, with the purpose of developing maximum company prestige, should include a definite policy of recognizing engineering achievement in releases to newspapers, magazines, radio and television.

In giving credit for group accomplishment it is of course necessary to exercise care, because many individuals may be involved in any one project and it may be difficult to single out for mention only one or two people without seeming to slight the others. But if the engineers in responsible charge are specifically mentioned and essential contributions of the others who worked with them are recognized, even anonymously, the morale of the whole group will be given a big lift and there will be an added incentive for each engineer to give of his best.

Properly told, the story of a new design and the men behind it can heighten public interest in both product and company, enhance the prestige of engineers, and attract urgently needed fresh blood into the profession. A corps of competent engineers is just about the biggest asset a manufacturer can have, and should be brought into the limelight, not buried in "dungeons of industrial oblivion."

Colin Carmichael
EDITOR

Fig. 1—Aircraft instrument utilizing a complex cast chassis. Critical space factor makes high accuracy a necessity



DIMENSIONING CASTINGS

Recommendations for correct detail specifications on casting drawings to insure feasibility, low cost and ease of manufacture

By Elvin V. Lundstedt

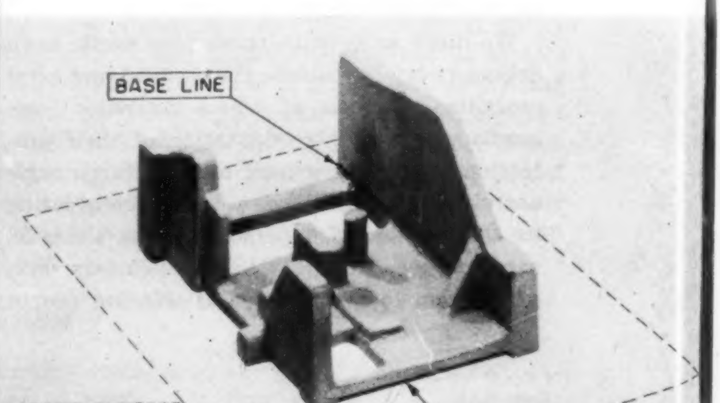
Meter and Instrument Dept., General Electric Co., Schenectady, N. Y.

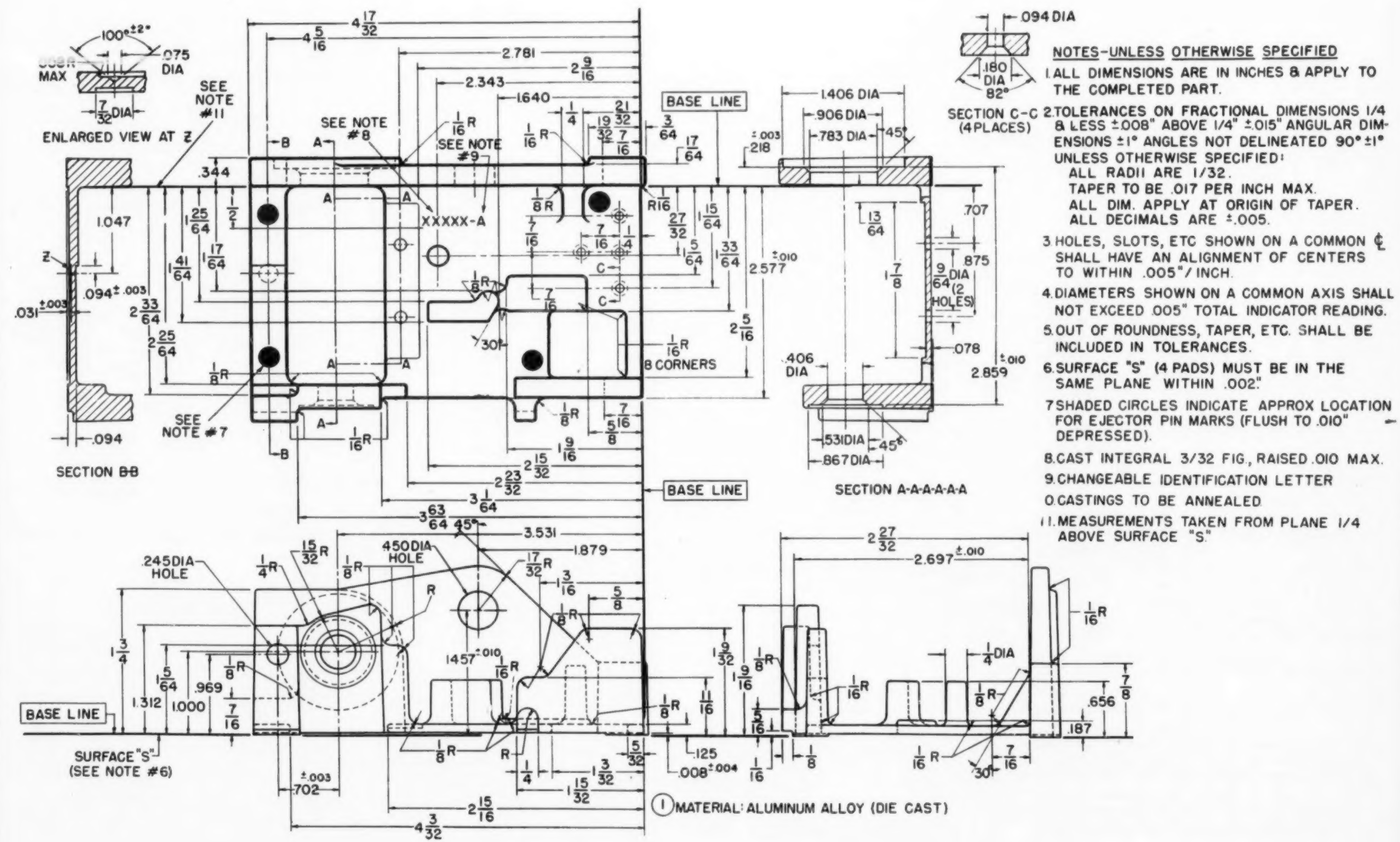
INCREASED complexity and higher accuracy of parts and dimensioning, as a result of technical advances in engineering, emphasize the need for good manufacturing drawings that show clearly all engineering requirements. Nowhere is this more true than in the case of complex castings such as those that form the basic structure of precision instruments and similar high-accuracy devices. A sound approach to dimensioning such castings which has proved successful in a wide variety of applications, including precision gyroscopes for aircraft and other measuring devices of a similar nature, *Fig.* 1, will be outlined in this article.

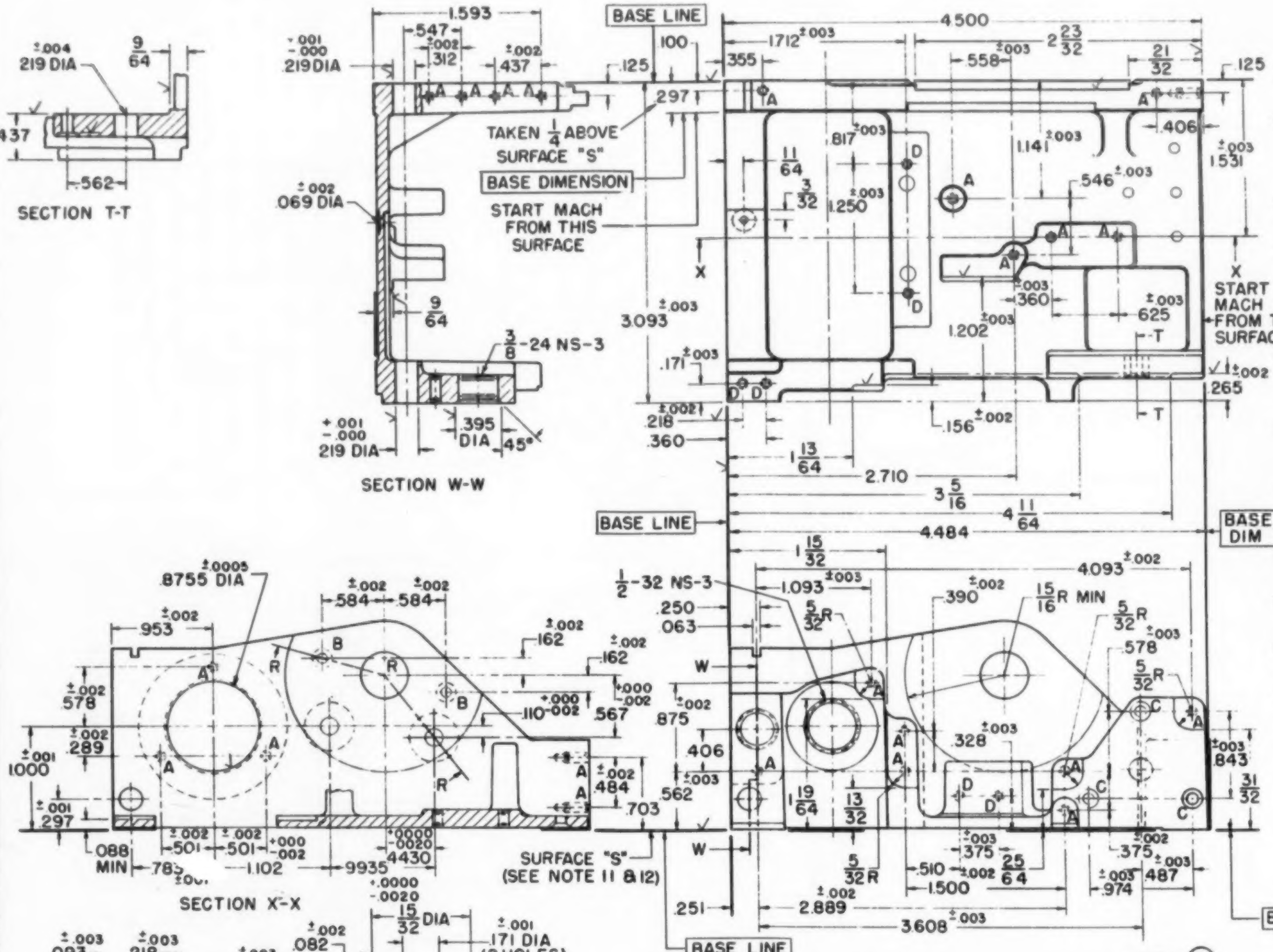
This material was prepared to facilitate dimensioning most efficiently a part to be produced by casting processes. It is not intended that this information replace the vitally important procedure of consulting those who are to produce the casting and perform the machining operations. The objective, rather, is to produce initial drawings of a practical nature that will require a minimum number of revisions when finally released for manufacturing. Use of this information must be tempered with discretion for each specific application. It is wide in its scope and actual limitations are dependent on the many interwoven requirements of each part.

Basic System of Drawing: There are many opinions on how to choose the most efficient method of producing drawings of cast items. There are three basic schools of thought—one is to make a drawing of the finished casting and rely on the pattern or moldmakers to allow an extra amount of material for machining purposes. For noncritical castings this is sufficient. A second approach is to make a composite drawing of the fin-

Fig. 2—Typical drawing for a production casting and casting produced. Three base lines are used




ENLARGED VIEW AT Z
SECTION B-B
SECTION C-C (4 PLACES)
SECTION A-A-A-A-A-A
BASE LINE
SURFACE "S" (SEE NOTE #6)
SEE NOTE #7
SEE NOTE #8
SEE NOTE #9
SEE NOTE #11
XXXXX-A
8 CORNERS
NOTES-UNLESS OTHERWISE SPECIFIED
1. ALL DIMENSIONS ARE IN INCHES & APPLY TO THE COMPLETED PART.
2. TOLERANCES ON FRACTIONAL DIMENSIONS 1/4 & LESS ±.008" ABOVE 1/4" ±.015" ANGULAR DIMENSIONS ±1° ANGLES NOT DELINEATED 90°±1° UNLESS OTHERWISE SPECIFIED:
ALL RADII ARE 1/32.
TAPER TO BE .017 PER INCH MAX.
ALL DIM. APPLY AT ORIGIN OF TAPER.
ALL DECIMALS ARE ±.005.
3. HOLES, SLOTS, ETC SHOWN ON A COMMON ℄ SHALL HAVE AN ALIGNMENT OF CENTERS TO WITHIN .005"/INCH.
4. DIAMETERS SHOWN ON A COMMON AXIS SHALL NOT EXCEED .005" TOTAL INDICATOR READING.
5. OUT OF ROUNDNESS, TAPER, ETC. SHALL BE INCLUDED IN TOLERANCES.
6. SURFACE "S" (4 PADS) MUST BE IN THE SAME PLANE WITHIN .002".
7. SHADED CIRCLES INDICATE APPROX LOCATION FOR EJECTOR PIN MARKS (FLUSH TO .010" DEPRESSED).
8. CAST INTEGRAL 3/32 FIG., RAISED .010 MAX.
9. CHANGEABLE IDENTIFICATION LETTER
10. CASTINGS TO BE ANNEALED
11. MEASUREMENTS TAKEN FROM PLANE 1/4 ABOVE SURFACE "S".
(1) MATERIAL: ALUMINUM ALLOY (DIE CAST)



NOTES-UNLESS OTHERWISE SPECIFIED

1. ALL DIMENSIONS ARE IN INCHES & APPLY TO THE COMPLETED PART.
2. TOLERANCES ON FRACTIONAL DIMENSIONS 1/4 & LESS ±.008. ABOVE 1/4 ±.015. ANGULAR DIMENSIONS ±1°. ANGLES NOT DELINEATED 90° ±1°. ALL DECIMALS ±.005
3. REMOVE BURRS ON MACHINED EDGES, .003 TO .010 R.
4. ALL MACHINED INSIDE CORNERS TO BE .005R MAX (UNDERCUT NOT PERMITTED).
5. HOLES, SLOTS, ETC SHOWN ON A COMMON ℄ SHALL HAVE AN ALIGNMENT OF CENTERS TO WITHIN .005 / INCH.
6. DIAMETERS SHOWN ON A COMMON AXIS SHALL NOT EXCEED .005 TOTAL INDICATOR READING.
7. ERRORS IN FLATNESS & PARALLELISM OF MACHINED SURFACES SHALL NOT EXCEED .003 / INCH.
8. SQUARENESS OR FACE RUNOUT OF MACHINED ENDS & SHOULDERS SHALL NOT EXCEED .005 / INCH OF R.
9. OUT OF ROUNDNESS, TAPER, ETC SHALL BE INCLUDED IN TOLERANCES.
10. ALL MACHINED SURFACES SHALL HAVE A MAX SURFACE ROUGHNESS OF 63.
11. SURFACE "S" (4 PADS) MUST BE IN THE SAME PLANE WITHIN .001
12. ANY OR ALL PADS (SURFACE "S") MAY BE AS CAST (UNMACHINED) FOR APPROX 50% OF AREA.

HOLE	DESCRIPTION	NO OF HOLES
A	#3-56 NF-3 7/32 DEEP MIN	22
B	$.096^{+.003}_{-.002}$ DIA & C'SINK $.180^{\pm.005}$ DIA X 82°	2
C	$.104^{+.004}_{-.002}$ DIA & SPOTFACE 3/16 DIA X 1/32 MAX DEEP	3
D	#2-64 NF-3 7/32 DEEP MIN	6

ished casting with the cast surfaces which are to be removed by machining, indicated by phantom lines. This is usually satisfactory for simple castings without too many intricate shapes and contours. Complicated castings with machining operations make this type of drawing unsatisfactory as it is difficult to distinguish the casting requirements from the machining requirements.

The third approach is to make separate drawings of the raw casting and the machined part, *Figs.* 2 and 3. This method offers the utmost in clarity, and is particularly effective for complex designs where interpretation of drawings is difficult at best, and successful manufacture is dependent on a sound method of dimensioning. It is this approach that is the basis for the dimensioning techniques described.

For economical redesign it is sometimes desirable to modify a basic casting by slightly different machining operations, at the same time maintaining the original requirements of the casting. When this is done each machined casting is assigned a new part number. This results in a series of machined castings, each with clearly defined requirements. These may be on the same drawing for simple parts requiring few dimensions but, in the case of large complex parts, should be on separate sheets. The casting drawing supplies the foundry and moldmaker with complete information on the part to be produced. The amount of material to be removed by machining is controlled on the casting drawing and is not subject to various interpretations by the pattern or moldmakers. With proper dimensioning, and with allowance for tolerances taken into account, the inspection department may reject the raw casting prior to any costly machining operations should a particular surface lack sufficient material for machining. The machining drawing will supply those interested with the complete machining requirements that manufacturing is to perform on the casting to produce the finished part. In summarizing, when separate drawings of the casting and machined product are used, neither is cluttered up or confused by information vital to the other.

The Base Line: The fundamentals of proper dimensioning are based on the principle that all solids are defined by dimensions in three planes. For good dimensioning, a base line should be used for each of the three planes of the part, both as cast and as machined. This is a line from which the majority of dimensions in one plane originate, the primary purpose being to avoid accumulation of tolerances through chain dimensions, *Fig.* 4. The base lines of the casting drawing should correspond to the surfaces from which initial machining is started in each plane. The base lines of the machining drawing should correspond to the surfaces from which subsequent machining is done, and must be related on the machining drawing to the base surfaces of the casting by a dimension which may be referred to as the base dimension.

The importance of correctly choosing the base

Fig. 3—Typical production machining drawing for the casting of *Fig.* 2. Three base lines and three base dimensions are used. Because the base surface of the casting and the machine drawing are common, the base dimension in this plane is zero. "Start machining from this surface" reference is given for each plane. Machined casting is shown below with base surfaces cross hatched

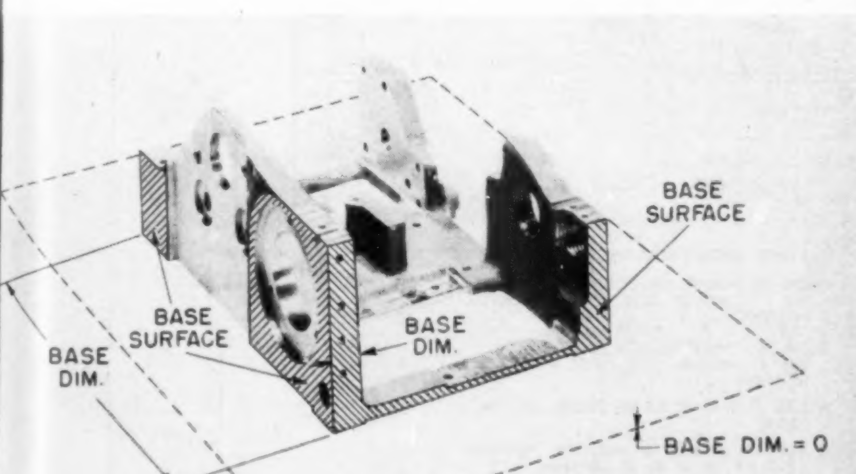
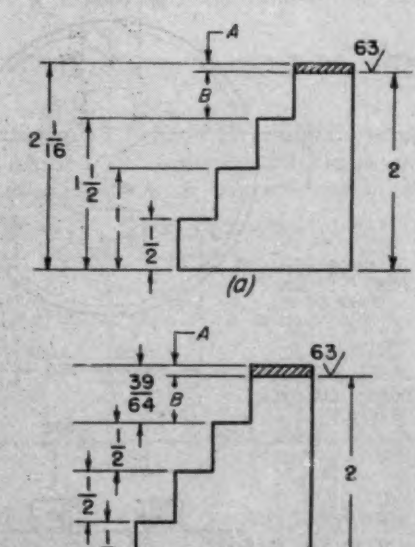


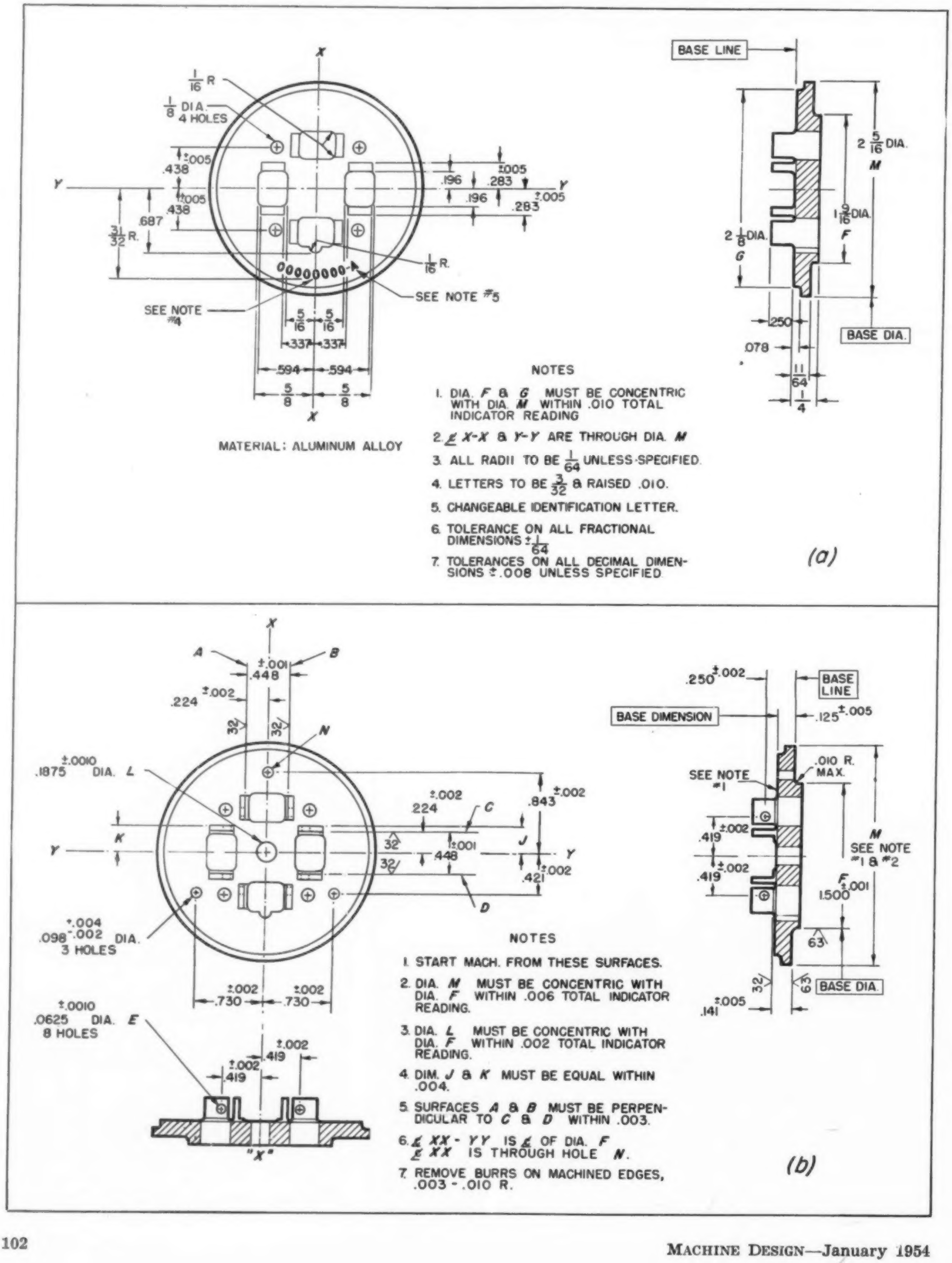
Fig. 4 — Typical base line dimensioning shown at *a* eliminates the accumulation of tolerances from chain dimensioning as at *b*



lines early in the design cannot be overemphasized, since a change in the base surface after the drawing is completed will mean many hours of re-calculating and re-dimensioning. The selection of these surfaces should be co-ordinated with both the people who are to perform the machining operations and those who are to produce the casting. A general guide for selection of these surfaces is as follows:

1. For the casting drawing, select surfaces that will not be removed by machining. Unless this rule is followed, control of the material to be

Fig. 5—Drawing for a casting with all surfaces properly referenced, *a*, and the final machining drawing for the casting, *b*, with all surfaces properly referenced



machined off is lost and inspection cannot determine the dimensions at fault when a surface fails to "clean up." When a casting is machined all over, some of these statements obviously must be violated. As an aid to tooling, a phantom line indicating the base surface and its base dimension may be indicated on the machining drawing.

2. Select surfaces that have good mounting features for jigs and fixtures, such as the three outermost points in one plane, so that any out-of-flatness would be minimized rather than exaggerated as it would be by mounting on points close together.
3. Select surfaces least subject to distortion when clamped because a part machined under stress will spring back when removed from a jig or fixture, resulting in extra dimensional variation.
4. Select surfaces integral with the main body of the casting, eg., not subject to the tolerance variations produced by cored surfaces, parting lines, or gated surfaces.

Base Line and Chain Dimensioning: A simple casting with a single machining operation is shown in *Fig.* 4 properly dimensioned at *a* and at *b* improperly dimensioned for this application. Note the amount of material to be removed by machining, dimension *A*, and the variation of the *B* dimension. The improved conditions of *Fig.* 4*a* are obtained by correct dimensioning, without increased cost or hardship to manufacturing. A comparison of dimensions *A* and *B* with both types of dimensioning will illustrate this point. Under the conditions that: (1) Tolerances on all fractional dimensions range plus or minus 1/64-inch, and (2) minimum stock for machining purposes is held to 1/32-inch, by calculating using the worst possible combination of dimensions and tolerances the following actual tolerances result.

	A	*B*
Fig. 4*a*	$\frac{1}{16}\pm\frac{1}{32}$	$\frac{1}{2}\pm\frac{1}{32}$
Fig. 4*b*	$\frac{7}{64}\pm\frac{5}{64}$	$\frac{1}{2}\pm\frac{1}{16}$

The Base Dimension: On the machining drawing a base dimension is a dimension from a cast surface to a machined surface. The base dimension, almost without exception, should be established between the base line on the casting drawing and the base line on the machining drawing and only one base dimension should be shown for each plane. A note should be added to flag this dimension by stating on the machining drawing "start machining from this surface", *Figs.* 3 and 5.

When the base surface of a casting is large in area, tapered, contoured, etc., it is sometimes desirable to add pads on the casting in order to insure that the jigs and fixtures will locate on specific areas. Another method is to indicate an area to which the base dimension and tolerance applies. This should be done in conjunction with those making the tools to insure that the proper surfaces are selected. The tolerance is then not subject to surface variations caused by any distortion of large cast areas. If the surface involved is located on three points of the casting by the tool and inspected on three arbitrarily chosen points, manufacturing would be penalized by the amount of distortion on each casting and its consequent resulting difficulty in holding the required tolerance, *Fig.* 6.

All dimensions on both casting and machining drawings should be from the base line. The only exception to this practice is on fitted parts where the actual location of the part is not critical, but the clearance of an integral portion is, *Fig.* 7.

The Base Diameter: Only one base dimension per axis should be shown on casting and machining drawings. The base diameter is the diameter to which all other diameters in the same plane are referenced. Base diameters of casting and machining drawings should be controlled by a concentricity requirement—which includes any out of roundness — stated on the machining drawing,

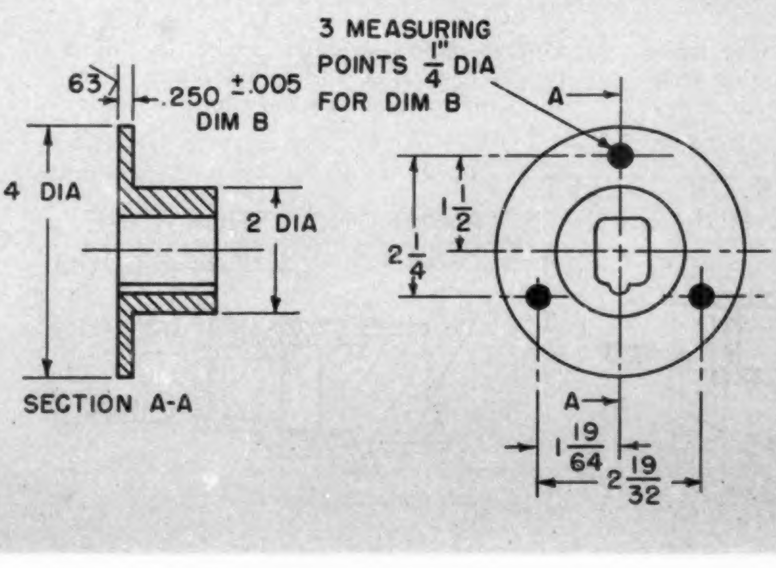


Fig. 6 — Left — Drawing showing method of applying tolerances to specific areas to control variations

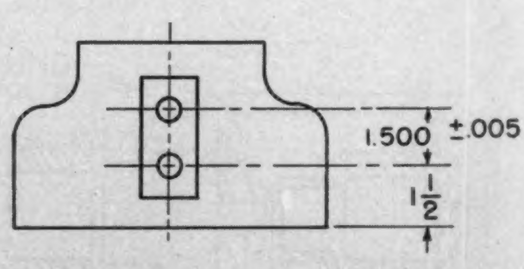


Fig. 7 — Above — Design showing the general case that is an exception to full base line dimensioning

Fig. 5*b*, in clearly understandable terms.

Identification of Centerlines: In order to avoid misinterpretation of centerline dimensions, centerlines should be identified as the axes of one diameter. This is accomplished by adding a note to the drawing indicating that centerlines *X-X* and *Y-Y* are through a designated diameter. This should be through the base diameters of both the casting and machining drawings. All dimensions in the *X-X* plane are referenced to centerline *X-X* and all dimensions in the *Y-Y* plane are referenced to centerline *Y-Y*. In *Fig.* 8 the correct practice is shown. The use of this note obviates common tooling troubles and varying interpretation of the drawing requirements. Many draftsmen would assume that centerlines on a drawing were the true centerlines of all diameters on a common axis. This is usually not true due to the improbability of manufacturing holding two or more diameters without some eccentricity.

The assembly in *Fig.* 8 illustrates typical case of machined mating parts where it is desirable to maintain proper clearance for a mounting screw. The method shown in *Fig.* 8*b* for dimensioning is preferred because hole *C* is identified relative to its design application and represents good tolerance utilization. It is also referenced to its functional diameter *B* in order to maintain proper clearance for the screw. The technique shown in *Fig.* 8*c* is undesirable on the basis that hole *C* is not identified with either diameter *A* or *B*. The eccentricity allowable by the note would result in no tolerance for dimension *D* when measured indiscriminately from diameter *A* or *B*.

Production Difficulties: Drawings should be dimensioned for harmony in product and tool requirements. The dimensions on the machining drawing should be selected in such a way that the tools used to produce the part will locate on one end of the dimension and perform the desired operation on the other end. Likewise, the tools used to produce the part should be designed in such a

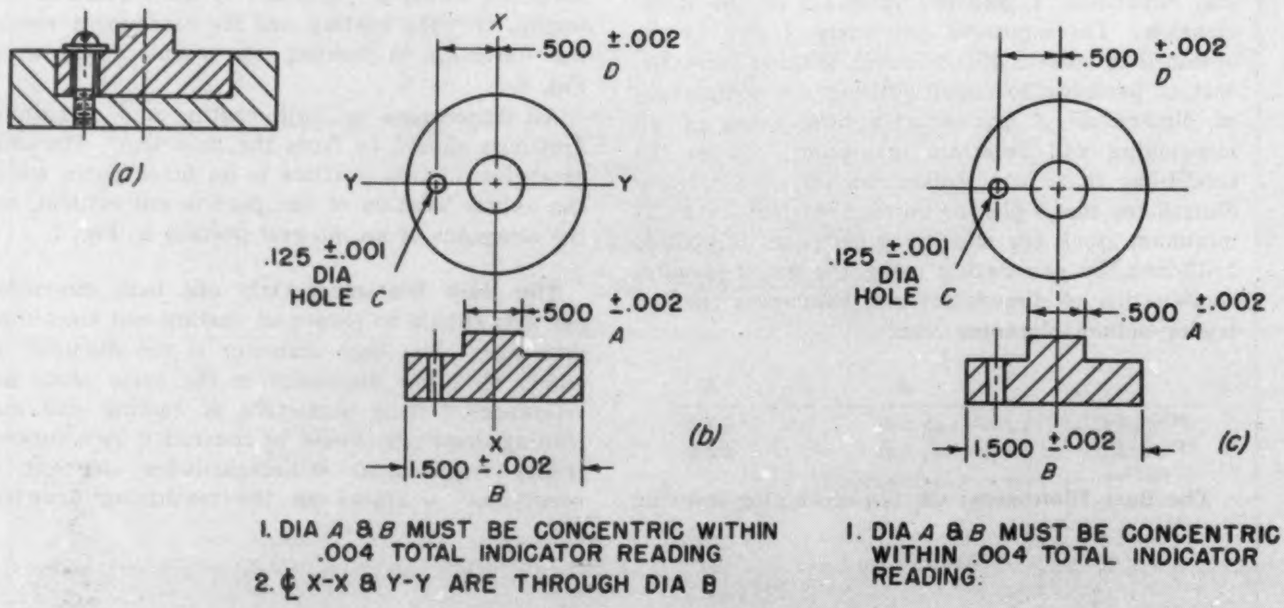
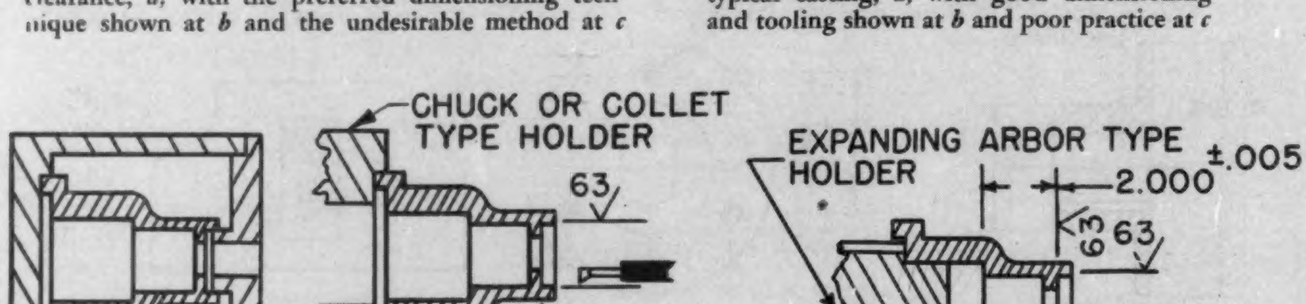


Fig. 8—Above—Design application involving screw clearance, *a*, with the preferred dimensioning technique shown at *b* and the undesirable method at *c*

Fig. 9—Below—Design application of a typical casting, *a*, with good dimensioning and tooling shown at *b* and poor practice at *c*



way that the part will be located on one end of the dimension and the desired machining operation performed on the other end, *Fig.* 9. The most common source of error and tolerance variation in machined castings, causing production difficulties and scrap, is encountered when a drawing has been dimensioned from one surface and the tool locates from some other surface. This has the same effect as chain dimensioning, that is, through accumulation of tolerances the part produced is beyond drawing tolerance. The only way to avoid this is by predetermining a tooling approach for each base dimension or base diameter on a drawing prior to final detailing of the casting drawings, *Fig.* 10. This example illustrates how properly dimensioned drawings, *Fig.* 5, may be planned for machining operations.

First Operation: Turn diameter *F*, face off and bore as required by *Fig.* 5. The initial machining is started from the surfaces indicated in Note 1 of *Fig.* 5*b* and with this type of setup, appropriate stops may be set on the machine and each part completely machined without changing the tool setting.

Second Operation: Drill three holes as required by *Fig.* 5*b*. Additional machining is done with the part locating on the base line and base diameter shown on the machine drawing. Also note that the part must be placed in the drill jig in the same position as the plan view because of the foolproof pin, *Fig.* 10*c*. This assures that the center lines *X-X* and *Y-Y* on the machine drawing are coincident with center lines *X-X* and *Y-Y* on the casting drawing.

Third Operation: Mill two slots 90 degrees apart as required by *Fig.* 5*b*. Additional machining is done with the part locating on the base line and the base diameter of the machine drawing. Also note the part must be placed in the milling fixture in the same position as the plan view of the machine drawing and consistent with the second operation (See foolproof pin). With this type of setup the milling operations can be performed on more than one part without changing the setting of the milling cutter.

Draft: Allowable draft should always be specified on the drawing. This information may be given in several forms, such as degrees, number of thousandths per inch, or as plus or minus a number of thousandths taper above and beyond the dimension and its tolerance. In critical applications, draft information may be covered by a note "taper to be included in tolerance unless otherwise specified" or "no taper allowable".

Fillets and Radii: Specify all fillets and radii on the drawing. Generous radii will strengthen castings, avoid stress concentrations, and aid in the flow of metal. This information may be expressed by designating each separate radius or, preferably, in note form by the selection of a common radius. This will result in economical castings and clear drawings, because in many applications the same cutting tool could be utilized if a common radius

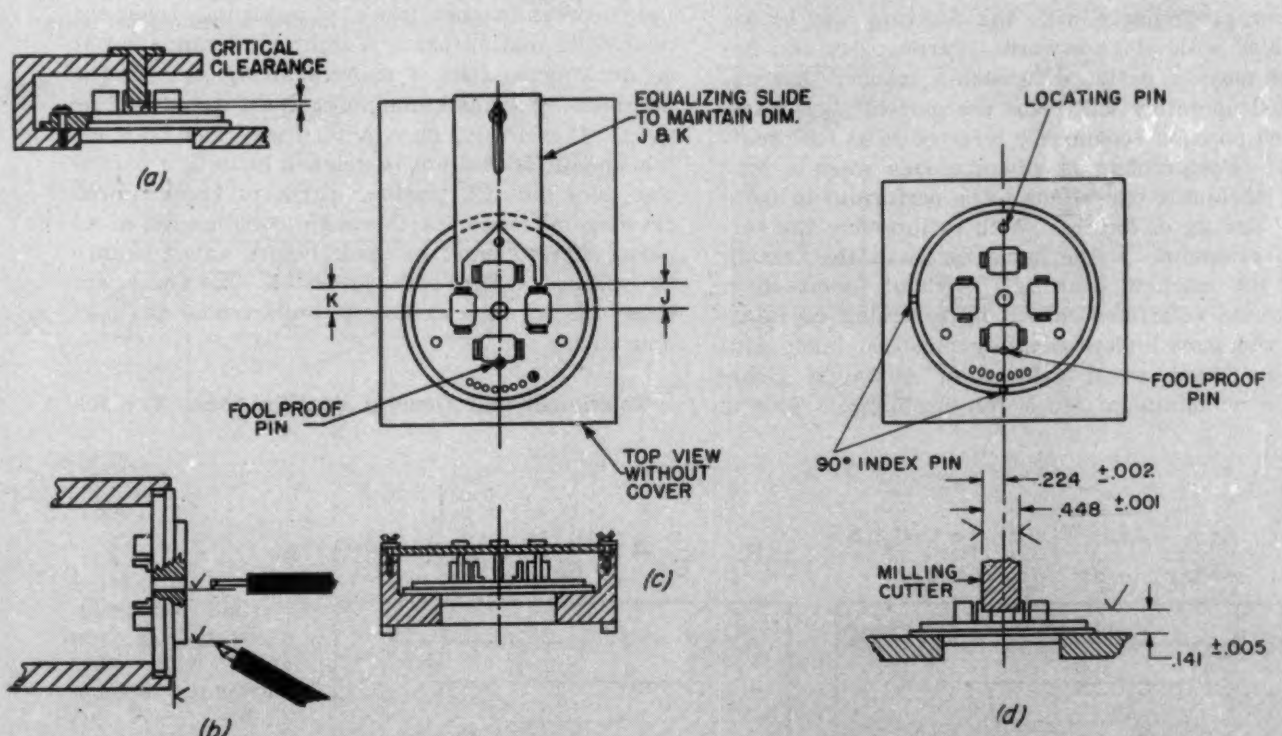


Fig. 10—Design application of roller plate part shown in *Fig.* 5 is indicated at *a* with first operation turning chuck or collet fixture at *b*, second operation drill jig at *c*, and third operation milling fixture at *d*

were selected; whereas, with a different radius, different tools would be necessary, *Fig.* 11.

Finishes: In order to maintain engineering control of functional appearance, or for other reasons, surface roughness symbols should be specified on the machining drawing. These will also identify surfaces to be machined. An example of the selection of roughness symbol is indicated in *Fig.* 3, the equivalent of *f*4, which is considered "run of the mill" machining. When most of the surfaces of a part are to be of the same roughness, this roughness should be specified by a note, "All machined surfaces to be μ63 unless otherwise specified". Other surfaces, more or less critical in their requirements, should carry their proper roughness symbols, *Fig.* 5.

Pattern Numbers: A drawing number and pattern letter should appear on the casting for identification purposes. In order that a finished casting may be identified, these markings should be placed on surfaces that are not to be machined. If this cannot be done it is still an asset, for stocking purposes and also for identifying castings that have pattern changes, to have this identification on the casting up to the time it is machined off. Should the pattern be revised this aids in replacing parts and also enables manufacturing to control the processing of the casting, *Fig.* 5.

Foolproofing: If a casting is symmetrical, the designer, in conjunction with manufacturing, should create an unsymmetrical surface, cutout, boss, etc. This will enable the part to be identified in its proper relationship with the drawing and be assembled without guesswork. Various jigs and fixtures may be designed in such a manner that unskilled operators will place the part in a predetermined position (commonly referred to as foolproofing). Foolproofing is advantageous when a part has machining operations to be performed in more than one jig or fixture. With foolproofing the vertical centerline is the same for both the casting and the machine drawing. Without foolproofing, tolerance variations caused by reversing or rotating the part indiscriminately must be taken into consideration when calculating minimum clearances or minimum stock for machining. This is especially true with ordinate dimensioning, *Fig.* 12.

Drill Spots: For noncritical applications, holes may be economically produced in cast parts by casting drill spots. These drill spots should have an included angle of 100 degrees plus or minus 2 degrees so that the body of the drill will be properly supported before the point begins to cut. This will minimize any additional tolerance caused by the drill not following the true center of the drill spot.

Dimensioning: Best practice is to dimension to actual surfaces in preference to intangible surfaces or centerlines. Holes and complete diameters are exceptions to this practice. In most instances this will enable the inspection department to use available tools rather than special time consuming setups, *Fig.* 13.

Tolerances: For obvious economical reasons it is best to use the most liberal tolerances that design requirements will allow. Actually, the cost of most manufacturing processes may be measured in terms of tolerances—this is also true in cast and machined items. The draftsman, designer and engineer should be able to justify economically the use of any tolerances tighter than plus or minus 0.015-inch. Unnecessary tolerances add hardship and scrap to the manufacturing burden. Tight tolerances should not be put on drawings because a particular manufacturing process is capable of producing parts to these tolerances, nor should they be placed on drawings with the "safety factor principle"—when a part needs a particular tolerance relative to mating parts, a tighter tolerance is put on drawing so that if manufacturing exceeds the tolerance by a small amount the part would still be good. Conversely, once a tolerance has been established it should not be relaxed because a particular tool did not produce parts to the required drawing tolerance. A thorough investigation of all parts affected must be made before any tolerance is loosened. The only justifiable tolerances are those which, when exceeded, would render the part unusable.

Tolerances and General Specifications: The fol-

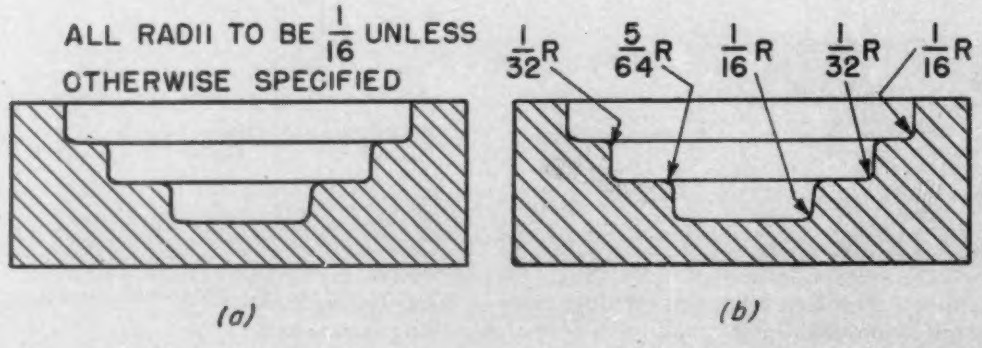


Fig. 11—Preferred method for designating radii is given at *a* while that shown at *b* is undesirable and less economical

lowing data show typical tolerance values for light-mass die, investment and sand castings. They are not intended to be restrictive; they are not the absolute minimum obtainable and should not be used if a broader tolerance is satisfactory for the application.

Die Casting Process: For die castings, such as those shown in *Fig.* 14, produced in aluminum the general tolerances can be specified as follows:

Linear: The preferred tolerance is plus or minus 0.010-inch up to 6 inches, over 6 inches allow an additional 0.002-inch per inch. However, plus or minum 0.0015-inch per inch may be held, but not less than plus or minus 0.002-inch for any dimension.

Angles: The preferred tolerance is plus or minus 1 degree, but in some instances plus or minus ¼ degree may be obtained.

Additional Tolerances: Additional tolerances of plus or minus 0.005-inch should be allowed for moving die parts. A minimum of 0.005-inch radius should be specified for threads and serrations.

Edges: An edge from 0.012 to 0.015-inch may be obtained provided a gradual taper is used and there is no abrupt section change.

Draft: The draft on diameters or cross-sectional dimensions on inner walls and cored holes should be 0.012-inch for the first inch of diameter plus 0.002-inch for each additional inch of diameter times the depth in inches. For example, with a 1½ inch diameter hole 1½ inches deep, total draft equals 0.012-inch plus ½ times 0.002, or 0.013-inch per inch of depth. The total draft would be 1½ times 0.013 or 0.020-inch. Outer walls should have a minimum draft of ½ degree per side. (0.009-inch per inch.)

Flatness: The flatness of a part should be calculated as 0.0015-inch times the length of the longest dimension.

Holes: Holes may be cast to a minimum of 0.100-inch diameter. A depth of four times the diameter should not be exceeded.

Wall Thickness: A preferred minimum wall thickness of 0.050 to 0.060-inch is desirable for small castings (less than 6 x 6 x 6 inches). Generous tapers should be utilized to join thick and thin sections.

Machining Allowances: After all dimensions involved have been calculated, using the worst possible combination of their tolerances, an additional 0.015-inch minimum should be allowed. A nominal amount of 1/32 to 3/64-inch is generally allowed for machining purposes.

Fillets: A fillet of 1/32-inch minimum radius is

Fig. 12—Flat plate design which includes a foolproof feature is shown at *a* while an undesirable design without is indicated at *b*

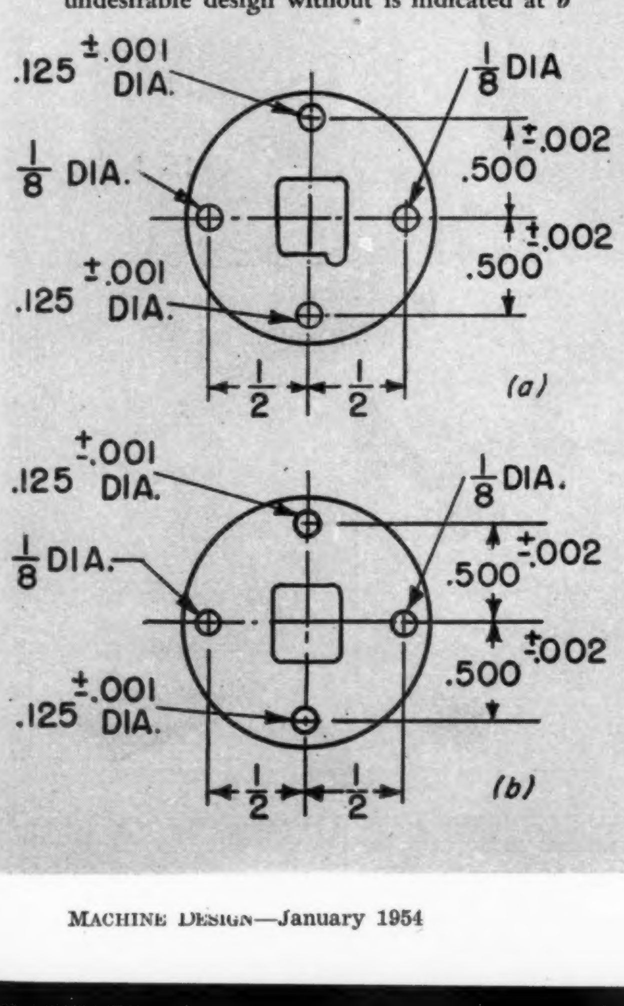
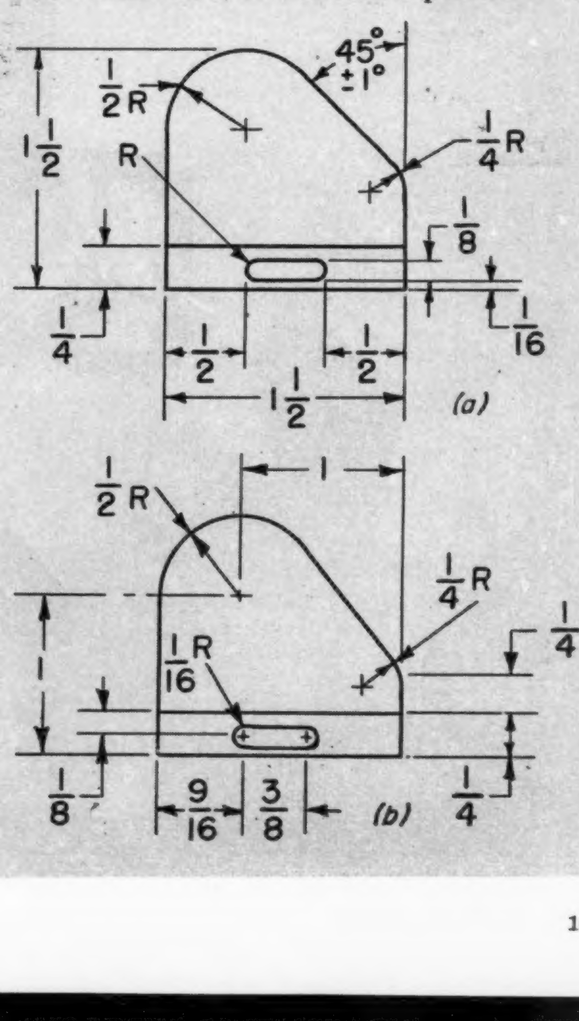


Fig. 13—Preferred dimensioning to tangible surfaces is shown at *a*. That at *b* is undesirable practice



desirable on all inside corners.

Ejector Pins: The permissible location and area for ejector pins should be indicated on drawings.

INVESTMENT CASTING PROCESS: For castings of the variety indicated by *Fig.* 15, cast in aluminum, brass, monel, or stainless steel, tolerances to observe are:

Size: The most practical application for this process is for one or more castings capable of fitting within a cylinder 5 inches in diameter and 6 inches long.

Linear: A preferred tolerance is plus or minus 0.005-inch per inch. However, on dimensions of ¾-inch or less, plus or minus 0.003-inch is allowed.

Angles: The preferred tolerance is plus or minus 1 degree, but in some instances plus or minus ¼ degree may be specified.

Additional Tolerance: An additional tolerance of plus or minus 0.005-inch should be allowed for dimensions across parting lines or gated surfaces.

Edges: An edge from 0.012 to 0.015-inch may be obtained provided a gradual taper is used and there is no abrupt section change.

Draft: In general, no draft is required but, in some instances, it is advantageous to have from ½ to 1 degree draft per sidewall. The drawings should specify those surfaces which functionally can or cannot have draft.

Flatness: Flatness should be calculated as 0.002-inch times the length of the longest dimension.

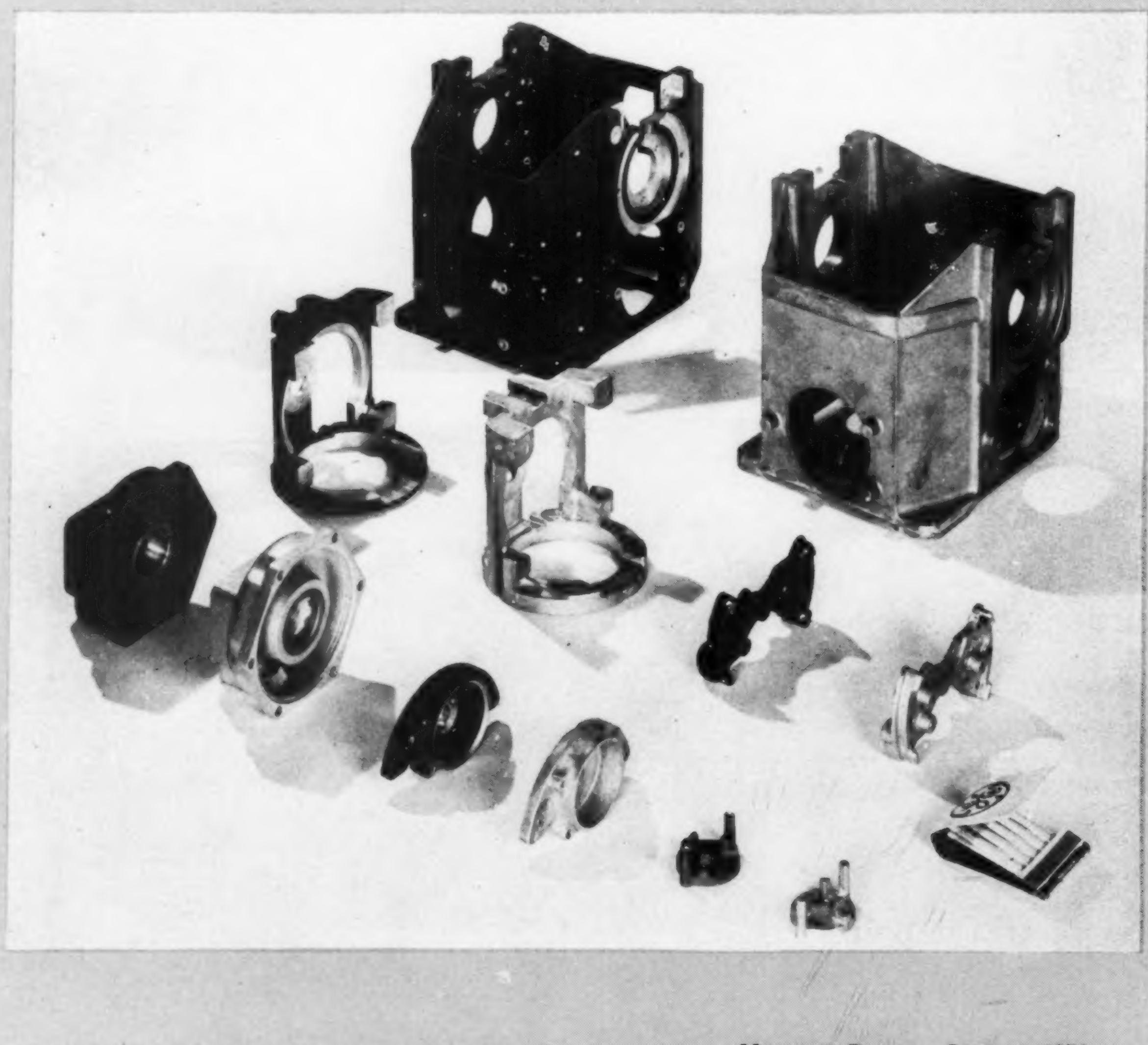
Holes: Holes may be cast to a minimum of 3/32-inch diameter. The depth of a blind hole should be equal to or less than the diameter. Through holes should not exceed a depth of three times the diameter.

Wall Thickness: A preferred minimum wall thickness of 0.050-inch is desirable. However 0.035-inch may be obtained on small areas. In all cases uniform walls are preferred. The thickest practical section is 1 inch. Generous tapers should be utilized to join thick and thin sections.

Fillets: Minimum fillet should be 1/64-inch radius and where possible 1/16 to ⅛-inch.

Machining Allowance: After all dimensions involved have been calculated using the worst possible combination of their tolerances, an additional

Fig. 14—A group of typical instrument diecastings. In each case the part shown on the left is machined



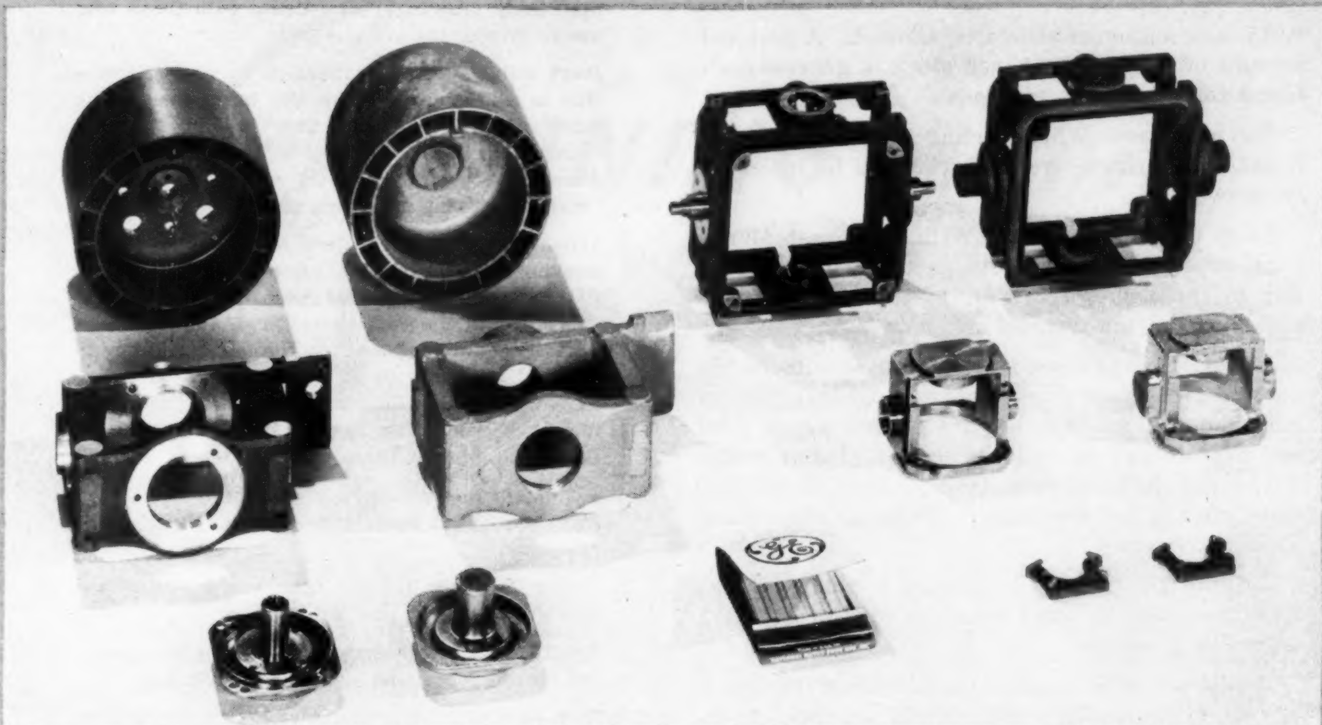
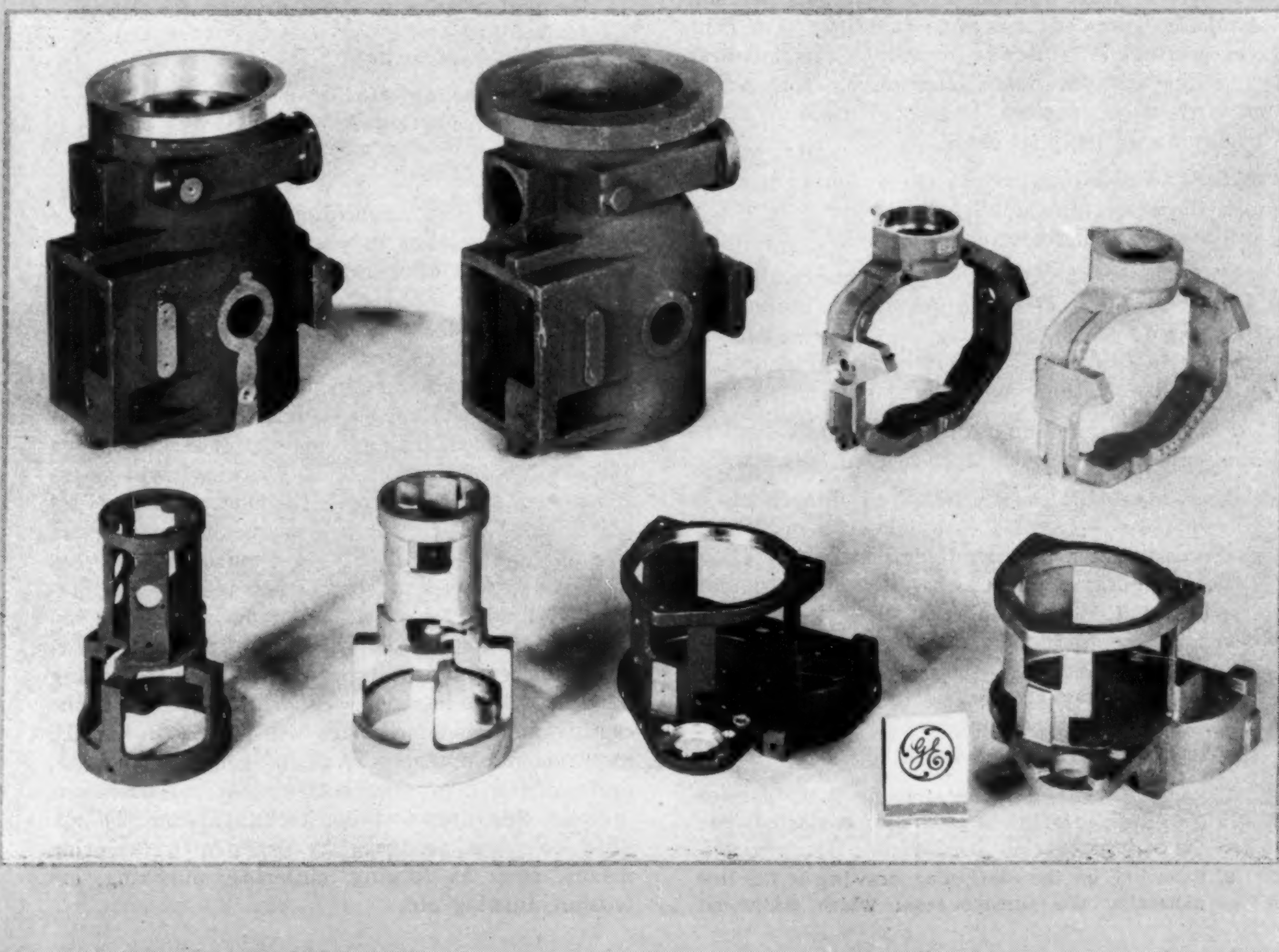


Fig. 15—Above—Typical instrument investment castings discussed showing the as-machined part on the left in each instance

Fig. 16 — Below — Sand castings employed in instruments. Machined castings are shown on left of each



DIMENSIONING CASTINGS

0.015-inch minimum should be allowed. A nominal amount of 1/32 to 3/64-inch stock is generally allowed for machining purposes.

Parting Lines: When parting lines are not permissible on critical areas, this should be indicated on drawing.

SAND CASTING PROCESS: With sand cast aluminum, brass and magnesium parts somewhat similar to those shown in *Fig.* 16 the general tolerances can be summarized as follows:

Linear: The preferred tolerances on dimensions up to 6 inches are plus or minus 1/32-inch, over 6 inches and up to 10 inches are plus or minus 1/16 and over 10 and up to 15 inches are plus or minus 3/32-inch. In general, as many dimensions as possible should be fractional. Decimal dimensions should be avoided.

Angles: The preferred tolerance is plus or minus 2 degrees, but in some instances plus or minus 1 degree may be held.

Additional Tolerances: An additional tolerance of plus or minus 1/64-inch should be added to dimensions produced by cored sections or across parting lines.

Edges: An edge from 1/32 to 3/64-inch may be obtained, provided a gradual taper is used and there is no abrupt section change.

Draft: A draft of from 1 to 2 degrees per side should be specified on simple castings. For complex castings it is usually desirable to submit first, drawings without draft, and obtain the recommendations of pattern and mold makers before specifying on the final drawings.

Holes: Holes may be cast to a minimum of ¼-inch diameter; the depth of a blind hole should not exceed its diameter.

Wall Thickness: A preferred minimum wall thickness of ⅛-inch is desirable, however 1/16-inch may be obtained on small areas. In all cases uniform walls are preferred. Generous tapers should be utilized to join thick and thin sections.

Fillets: Minimum fillet should be 1/32-inch radius and where possible 1/16-inch to ⅛-inch.

Machining Allowance: After all dimensions involved have been calculated using the worst possible combination of their tolerances, an additional 1/32-inch minimum should be allowed. A nominal amount of 1/16-inch finish stock is generally allowed for machining purposes.

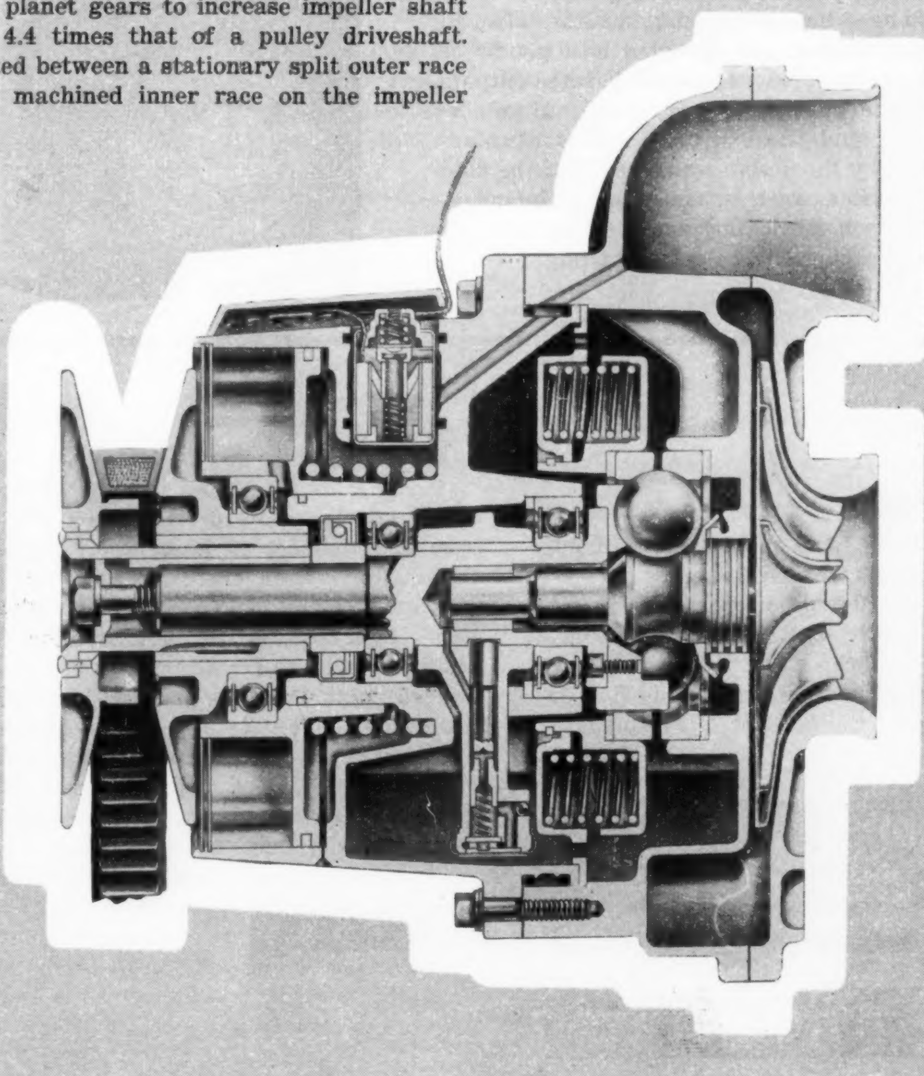
General Rules: To summarize, the overall rules to observe in dimensioning castings for most effective results are then as follows:

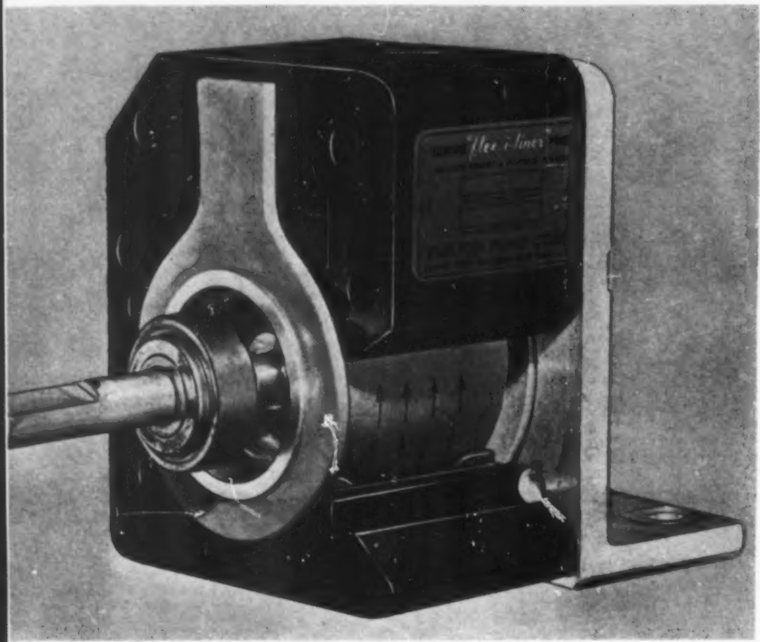
1. Base line on the casting drawing is the surface from which the initial machining is started (one for each plane).
2. Base line on the machining drawing is the line indicating the surface from which additional machining is started (one for each plane).
3. In selecting a base line for the casting and the machining drawings, co-ordinate with those who are to provide the various tools.
4. Have only one base dimension in each plane—this is a dimension from the base line on the casting drawing to the base line on the machining drawing. Add note to identify the base dimension by stating on the machining drawing "start machining from this surface".
5. When there are diameters on a casting to be machined, the base dimension is replaced by a note which connects the base diameter of the casting with the base diameter of the machined part. This note is placed on the machining drawing and is expressed by a concentricity requirement in total indicator reading. When dimensioning from centerline, identify the centerlines as being the axes of these base diameters.
6. All dimensions on casting and machining drawings should be from the base line (except on fitted parts).
7. Never machine off the base surface that was established on the casting drawing.
8. Always indicate with the proper roughness symbol the surfaces that are to be machined.
9. Use as many fractional dimensions as possible—be able to justify the use of any toleraces tighter than $\pm \frac{1}{64}$-inch.
10. Dimension to actual surfaces rather than to centerlines—except for complete diameters.
11. If the casting is symmetrical, create an unsymmetrical surface, cutout, boss, etc., to orient the part relative to detail drawing for layout, tooling and inspection purposes when critical clearances are involved.
12. Include casting drawing number and pattern letter for identification purposes. Place on surface that will not be machined.

Conclusion: The application of these principles in making drawings of cast items will result in many important advantages. Complicated casting drawings can be made efficiently by draftsmen with but a few years' experience. Drawings will not be subject to varying interpretations, by drafting, engineering, manufacturing, inspection and others who use them. Personal opinion will be replaced by clearly indicated requirements. Initial drawings will require fewer revisions. Drawings will be dimensioned for harmony of product and tool requirements. For proper design application and clearance purposes all cast and machined surfaces will be referenced to some other relative surface. Cost of final product will be reduced because, by proper selection of tolerances on each detail drawing, unnecessary scrap and other manufacturing burdens will be lessened. Separate drawings of the casting and machined part will clearly define the requirements of each. And, of equal importance, draftsmen will produce better all-round drawings, because this dimensioning technique may be applied to parts produced by other manufacturing means, such as forging, sintering, punching, extrusion, turning etc.

SCANNING the field for IDEAS

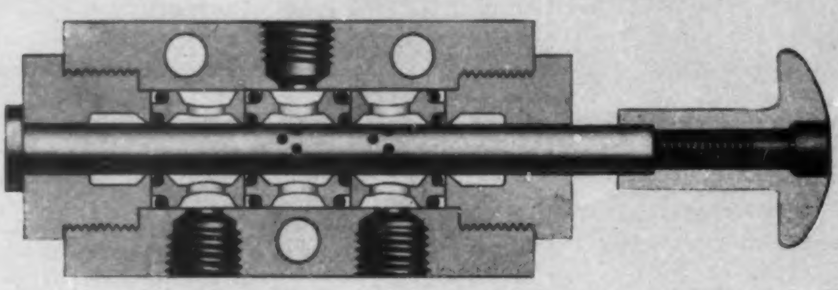
PLANETARY BALL DRIVE for stepping-up shaft speeds cuts down transmission noise and compensates automatically for effects of wear. Developed by McCulloch Motors Corp. for a new automotive supercharger design, the drive employs spring-loaded balls instead of conventional planet gears to increase impeller shaft speed 4.4 times that of a pulley driveshaft. Mounted between a stationary split outer race and a machined inner race on the impeller shaft, the balls are driven by arms rigidly fastened to the pulley shaft. Constant "meshing" of the balls is assured by coil springs which preload the split outer race against the inner, automatically taking up any wear and maintaining quiet operation.

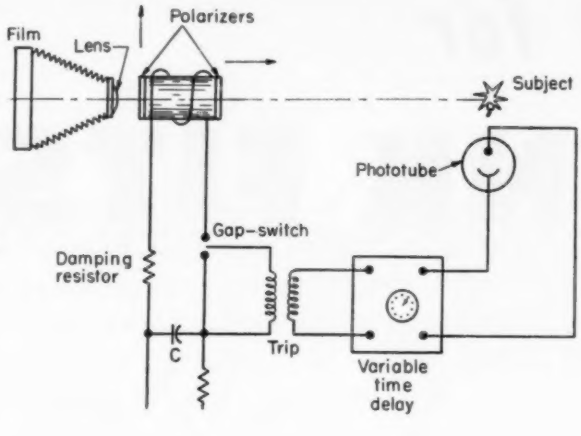


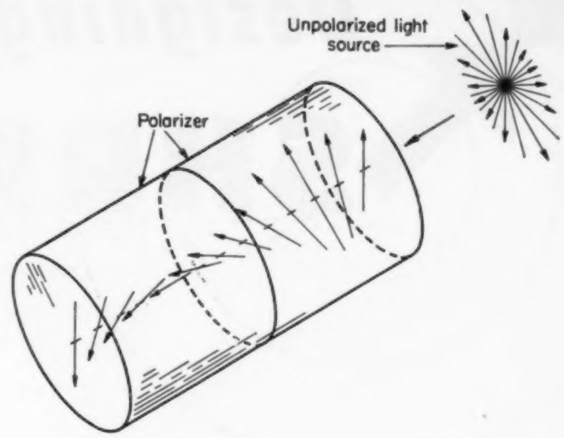


ELASTIC FLOW CONTROL in a pump designed for corrosive or abrasive fluids eliminates conventional gaskets or check valves and prevents fluid-to-metal contact. In the Vanton "flex-i-liner" pump, a rubber or synthetic flanged liner actuated by a rotor, offset to provide pumping action, and mounted in a plastic or hard rubber body is employed to provide rated outputs up to 20 gpm and pressures up to 60 psi. High corrosion resistance and freedom from fluid contamination are assured by the design which seals off the fluid passage from contact with moving parts. In operation, the liner is self-lubricating and the flexing action provides a self-priming effect during starting.

TUBULAR VALVE PLUNGER without lands or other surface irregularities reduces wear and facilitates production of a control valve developed by C. B. Hunt and Son Inc. The design has been simplified by a unique porting arrangement in which fluid flow is guided through the interior of the plunger from one chamber to another. Tendency of parts to creep or crawl during operation is eliminated by the construction which maintains a balanced force distribution. In addition, metal-to-metal seating has been avoided, minimizing friction and improving sealing effectiveness. Suitable for air, hydraulic or vacuum applications, the design can handle pressures up to 125 psi at temperatures up to 150 F.



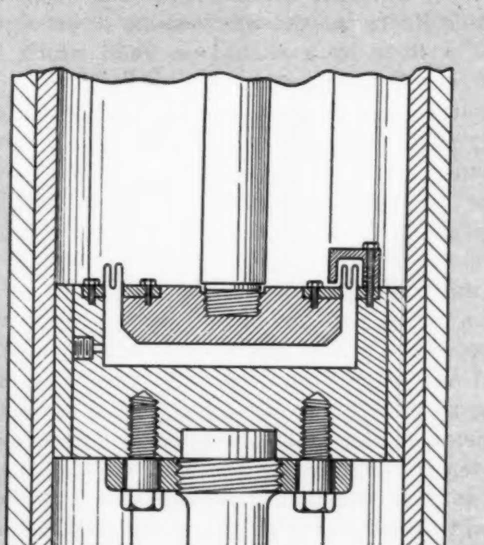




ONE-MILLIONTH SECOND SHUTTER ACTION is attained with a unique light polarizing shutter which has no moving parts. Developed by Edgerton, Germeshausen & Grier Inc. for the Atomic Energy Commission, the Rapatronic camera operates on the principle of the Faraday effect in which a magnetic field is employed to rotate a beam of polarized light and provide instantaneous exposures of about 0.8 microsecond in duration.

In operation, the shutter is triggered by a photoelectric cell. Light pulses, either from the subject or a separate flash unit, generate a signal to release the energy in a high-voltage condenser-discharge circuit, producing a short-duration high-peak magnetic field around two crossed polarizers which normally block light from entering the camera. Under the influence of the magnetic field, the plane of polarization is rotated, effectively "opening" the shutter. For high intensities of illumination, a third polarizer is frequently required to reduce passage of light with the shutter closed; however, this addition increases power requirements four times and necessitates a larger power supply. A high-quality photographic image is produced by the design which has been applied successfully to the study of nuclear detonations.

FLOATING FLUID JOINT for connection of reciprocating members operating under high loads simplifies assembly and minimizes troublesome effects of angular and lateral misalignment. In a design developed by Byron H. Leonard for high-pressure pumps, a fluid cushion is employed to transmit power from the driving crosshead to the piston rod on the compression stroke. Misalignment is absorbed by a bellows-type diaphragm which connects a plug on the end of the piston rod with the crosshead and acts as the fluid seal. To relieve strain on the diaphragm during the return stroke, the rod is driven by means of rigid fingers or channels mounted to the crosshead. Pressure developed in the fluid chamber which determines the necessary wall thickness and flexibility of the diaphragm, can be varied by changing the size of the plug on the piston rod. Compared to conventional mechanical connections, the design eliminates stiffness due to joint friction and reduces possibility of damage to packings.



Pat. 2,639,172

Designing for SHRINK FITS

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THE FUNDAMENTAL basis for shrink fitting, as for any prestressing method, is substituting favorably distributed stress for weight of material and thus increasing the effective range of elastic operation of a machine or structural part. The intent may be to increase the elastic range of operation of a pressure vessel, to increase the fatigue life of a wheel and axle assembly, or perhaps to strengthen an extrusion die. In many instances a low-cost material, weak from the standpoint of inherent strength, can be used in a prestressed condition to accomplish the same results as a more expensive, higher strength material.

Shrink-fit constructions are well suited to a number of different design situations, such as replaceable liners in pressure vessels, or strengthening of a liner by a shrunk-on shell which helps retard crack propagation. Its versatility is also demonstrated in assemblies composed of materials having different properties or degrees of corrosion resistance, such as a shrunk-on steel shell over a copper or aluminum liner, in which maximum utilization is made of the properties of both materials.

In this article, equations for calculation of principal stresses and shrinkage allowance, in which the geometry of the assembly is arbitrarily determined and stresses calculated on the basis of this geometry and the properties of the materials, will be reviewed. In addition, however, methods will be given for determining the best (optimum) geometry so that maximum performance may be expected with minimum weight of material. Examples are also given of such design methods applied to two-shell shrink-fit type pressure vessels.

Basic Theory: Fundamentals of shrink-fit theory are covered by analyses of heavy wall cylinders subject to internal and external pressure.[1-4] Fabrication procedures involved, lubrication of shrink-fit assemblies, preparation of contact surfaces and similar manufacturing problems[1-3] will not be considered here.

Cross sections of various types of shrink-fit construction, *Fig.* 1, are (1) hub and solid shaft under shrink-fit pressure only, *Fig* 1*a,* (2) hub and hollow shaft under shrink-fit pressure only, *Fig.* 1*b,* (3) two-shell vessel with internal pressure and shrink-fit pressure, *Fig.* 1b, and (4) hub and shaft subjected to shrink-fit pressure and centrifugal forces, *Figs.* 1*c* and 1*d*. In all cases the component parts are not necessarily of the same material.

In producing a shrink-fit type of construction, it is common to heat the outer component in order to expand it perhaps 0.003-inch beyond the interference on the diameter, and then slip it over the inner component while expanded. The temperature differential through which the outer part must be heated in order to obtain the required expansion is (see Nomenclature)

$$t_f - t_i = \frac{d_f - d_i}{\alpha\, d_i} \qquad (1)$$

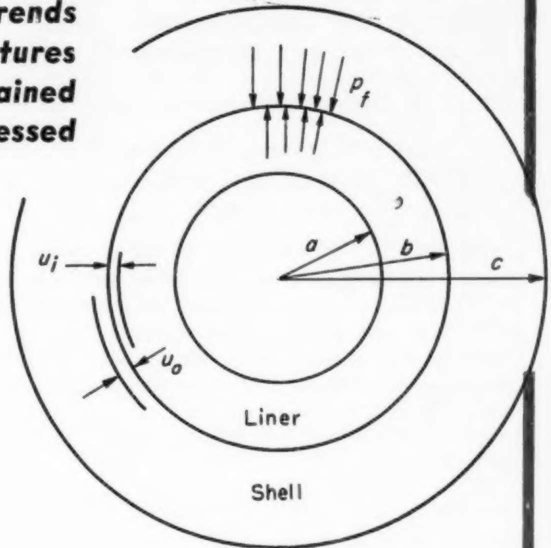
To increase the temperature differential or decrease the high temperature of the outer component, the inner component of the assembly may also be cooled so that it contracts. If cooling only is used then, instead of Equation 1,

$$-(t_i - t_f) = \frac{d_f - d_i}{\alpha\, d_i} \qquad (2)$$

The approximate size change, per inch of length or diameter, which occurs when parts are heated or cooled is shown on *Fig.* 2.[3] To determine size change for a given material on heating or cooling, start on *Fig.* 2 at the temperature specified, follow across the graph horizontally to the material in question, then follow the vertical lines to read the size change per inch of dimension. The size change

1. References are tabulated at end of article.

How increased efficiency of structural and machine parts—particularly important with present trends toward higher operating pressures, temperatures and speed—can be more economically obtained by using materials in their optimum stressed condition.



Nomenclature

a = Bore radius of hollow cylinder, inches
b = Interface radius or outside radius of inner component of two-shell shrink-fit assembly, inches
c = Outside radius of outer component of two-shell shrink-fit assembly, inches
d_f = Final diameter of cylinder, inches
d_i = Initial diameter of cylinder, inches
E = Modulus of elasticity when $E_i = E_o$, psi
E_i = Modulus of elasticity of inner component of two-shell constructions, psi
E_o = Modulus of elasticity of outer component of two-shell construction, psi
F = Axial force, pounds
f = Coefficient of friction between shell and liner
f' = Torsional coefficient of friction
g = Acceleration due to gravity, in./second^2
K = Constant (Equation 14)
= $(b^2 + c^2)/E_o\,(c^2 - b^2) + \mu_o/E_o + (a^2 + b^2)/E_i\,(b^2 - a^2) - \mu_i/E_i$
L = Length of fit, inches
N = Constant (Equations 58-60)
= $(1 - \mu^2)\gamma\,\omega^2/gE$
p_f = Shrink-fit pressure, psi
$_op_f$ = Optimum shrink-fit pressure, psi
p_i = Internal pressure, psi
$_mp_i$ = Maximum internal pressure, psi
$_op_i$ = Optimum internal pressure, psi
p_o = External pressure, psi
R = Wall ratio, c/a
R_1 = Wall ratio, b/a
R_{ob} = Ratio outside to bore radius for monobloc cylinders
r = Variable radius in cylinder, inches
s_h = Hoop stress in cylinder, psi
s_{hi} = Hoop stress in inner component of two-shell shrink fit vessel, psi
s_{ho} = Hoop stress in outer component of two-shell shrink-fit vessel, psi
s_i = Tensile yield strength of inner component of two-shell vessel, psi
s_o = Tensile yield strength of outer component of two-shell vessel, psi
s_r = Radial stress in cylinder, psi
s_{ri} = Radial stress in inner component of two-shell shrink-fit vessel, psi
s_{ro} = Radial stress in outer component of two-shell shrink-fit vessel, psi
s_{si} = Shear stress in inner component of two-shell shrink-fit vessel, psi
s_s = Shear stress when $s_{ri} = s_{ro}$, psi
s_{so} = Shear stress in outer component of two-shell shrink-fit vessel, psi
s_y = Tensile yield strength where $s_i = s_o$, psi
T = Torque, lb-in.
t_f = Final temperature, F
t_i = Initial temperature, F
u = Radial deformation of cylinder, inches
u_i = Radial deformation of inner component of two-shell vessel, inches
u_o = Radial deformation of outer component of two-shell vessel, inches
u_{rb} = Radial deformation when $r = b$, inches
V = Linear velocity at $r = c$, in./second
= ω/c
α = Coefficient of thermal expansion, in./in./deg F
γ = Density, lb/in.^{3}
Δ = Radial shrinkage, inches
$_o\Delta$ = Optimum radial shrinkage, inches
Δ' = Change in radial shrinkage
μ = Poisson's ratio
μ_i = Poisson's ratio for inner component of two-shell construction
μ_o = Poisson's ratio for outer component of two-shell construction
π = 3.1416
ϕ = s_o/s_i
ω = Angular velocity, radians/second

per inch multiplied by the length of diameter of the part gives the total size change.

EXAMPLE 1:[3] A 6-inch diameter steel shaft is to be cooled from room temperature to −310 F. Start at −310 F, follow horizontally across to the line for steel, then down to read approximately −0.0018-in./in. Multiply this size change per inch by the diameter of the shaft. The result, −0.0018 × 6 = −0.0108-inch, which is the total size change cooled from room temperature. To determine the diameter of this shaft at −310 F, subtract the total size change from the diameter at room temperature. The result, 6.0 −0.0108 = 5.9892 inch, is the diameter at the assembly temperature.

Fig. 2 may also be used to determine the temperature to which a part must be heated or cooled to effect a specified size change.

EXAMPLE 2:[3] To shrink a 5-inch diameter steel shaft 0.005-inch for assembly. Dividing the total shrink required by the diameter, the shrinkage must be 0.005/5 = 0.001-in./in. of diameter. Using the curve in *Fig.* 2, start at −0.001 inch, and follow up to the line for steel, then across to the temperature scale. The chart shows that the part must be cooled from room temperature to approximately −150 F.

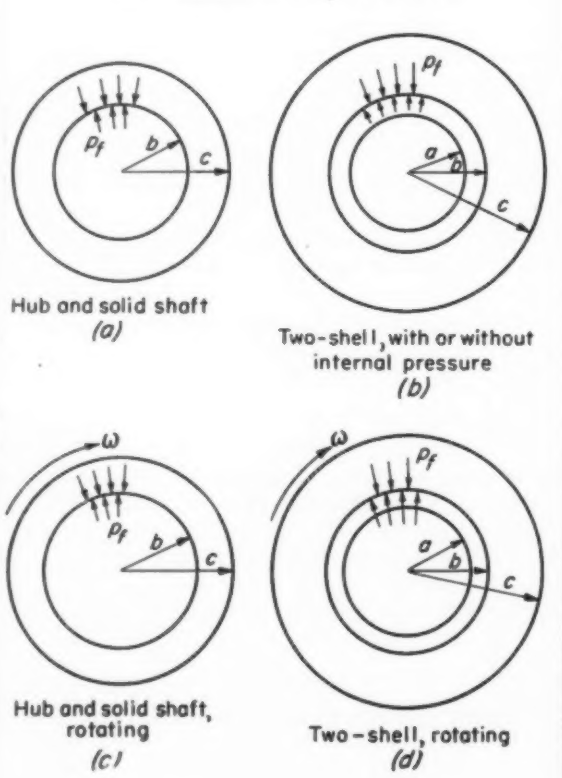
For those materials not covered in *Fig.* 2, Equations 1 and 2 may be used.

EXAMPLE 3:[3] Assume a ring of 10 inch inside diameter of a material having a linear coefficient of thermal expansion of 0.000007-in./in./deg F. This ring must be heated from room temperature (68 F) to 500 F to make a shrink assembly. By Equation 1

$$d_f = \alpha\, d_i (t_f - t_i) + d_i$$
$$= 0.000007(10)(500 - 68) + 10$$
$$= 10.03024 \text{ inch}$$

which is the inside diameter of the ring when heated.

Fig 1—Cross sections of various shrink-fit constructions



Some physical and mechanical properties of common materials that may be encountered in making shrink-fits are listed in TABLE 1. In this table only properties are included for annealed material, in which it is assumed that no residual stresses are present which would influence shrink-fit calculations. Steels tempered to 1000 F or higher usually contain fairly low heat-treatment residual stresses. However, caution must be exercised in noting mechanical properties because, as shown on *Fig.* 3, properties vary with section size. Consequently, for heat-treated materials particularly, the properties noted should be in reference to the size of the part involved.

As a result of the shrinking operation, stresses are induced in the component parts from the shrink-fit pressure generated at the interface. As shown in *Fig.* 1*b*, for example, the effect is such that the outer member is subjected to internal pressure and the inner member to external pressure. The principal stresses for a cylinder under internal and external pressure are[4]

$$s_h = \frac{a^2 p_i - c^2 p_o}{c^2 - a^2} + \frac{(p_i - p_o)\, a^2 c^2}{r^2 (c^2 - a^2)} \quad \text{(3)}$$

and

$$s_r = \frac{a^2 p_i - c^2 p_o}{c^2 - a^2} - \frac{(p_i - p_o)\, a^2 c^2}{r^2 (c^2 - a^2)} \quad \text{(4)}$$

For the inner component, which is under external

Table 1—Properties of Common Materials
(Annealed)*

Material	Tensile Strength (1000 psi)	Yield Strength (1000 psi)	Modulus of Elasticity (million psi)
Commercial bronze (90 Cu, 10 Zn)	37.5	11.5	17
Commercially pure copper	31.5	10	16
Wrought monel	75	35	26
Commercially pure nickel	70	20	30
2S-0 aluminum	13	5	10
52S-0 aluminum	29	14	10.2
Red brass (85 Cu, 15 Zn)	40	15	15
Yellow brass (65 Cu, 35 Zn)	48	18	14
Aluminum bronze (92 Cu, 8 Al)	70	25	15
Beryllium copper	70	30	18
SAE 1020 steel	60	40	30
SAE 1040 steel	79	48	30
SAE 1137 steel	85	50	30
SAE 1144 steel	98	52	30
SAE 2330 steel	95	59	30
SAE 8630 steel	90	60	30
SAE 4340 steel	109	67	30

*An approximation for Poisson's ration is $\nu = (E/2G) - 1$ where G is the shear modulus of elasticity. Coefficient of thermal expansion varies depending on temperature and can be found in standard handbooks.

pressure, the principal stresses by Equations 3 and 4 become

$$s_{hi} = \frac{-p_f\, b^2}{b^2 - a^2}\left(1 + \frac{a^2}{r^2}\right) \quad \text{(5)}$$

$$s_{ri} = \frac{-p_f\, b^2}{b^2 - a^2}\left(1 - \frac{a^2}{r^2}\right) \quad \text{(6)}$$

If the inner component is a solid shaft then $a = 0$ and

$$s_{hi} = s_{ri} = -\, p_f \quad \text{(7)}$$

that is, the hoop and radial stresses are uniformly compressive throughout the shaft and equal in magnitude to the shrink-fit pressure.

The outer component is under internal pressure, and the principal stresses according to Equations 3 and 4 are

$$s_{ho} = \frac{p_f\, b^2}{c^2 - b^2}\left(1 + \frac{c^2}{r^2}\right) \quad \text{(8)}$$

$$s_{ro} = \frac{p_f\, b^2}{c^2 - b^2}\left(1 - \frac{c^2}{r^2}\right) \quad \text{(9)}$$

The various stress distributions obtained are shown in *Fig.* 4.

In order to calculate the shrinkage, the amount of radial deformation involved, must be known. This deformation is

$$u = \frac{1-\mu}{E}\left(\frac{a^2 p_i - c^2 p_o}{c^2 - a^2}\right) r + \frac{1+\mu}{E}\left(\frac{a^2 c^2}{r}\right)\left(\frac{p_i - p_o}{c^2 - a^2}\right) \quad \text{(10)}$$

For a composite cylinder, Equation 10 is applicable to both parts; p_i and p_o represent the shrink-fit pressure, for example, and b would be substituted for c for the inner component, etc. The shrinkage allowance is the total deformation involved and is equal to the absolute sum of the individual deformations of the inner and outer parts, that is, the shrinkage allowance Δ is

$$\Delta = |\, u_i + u_o \,| \quad \text{(11)}$$

Before considering practical design cases and calculations, a few cautions in using the shrink-fit type of construction should be mentioned. First, it is necessary to know exactly the initial stresses in the components before assembly. If high residual stresses due to heat-treatment are present, for example, they must be taken into account in the design equations; otherwise the stresses and deformations obtained will not be in accordance with those calculated.[5] Shrinkage temperatures over 500 F may have an annealing effect on certain steels tempered to a high hardness level. High thermal gradients at the instant of contact of the components may induce high thermal stress and accompanying plastic flow if the stresses are above the yield strength of the material. Freezing of shells on liners or shafts before complete assembly is also an inconvenience frequently encountered.

Dimensional tolerances must be accurately controlled because stresses set up in shrink fitting are directly dependent on the accuracy of machining.

Calculation of Shrink-Fit Assemblies

Design for Shrink-Fit Pressure Only: Simplest case to be considered is when shrink-fit pressure only is taken into account; the effects of internal pressure and centrifugal forces are not present. Four general situations are covered.

1. Different Materials—Hollow Shaft: In the general case, the assembly is a two-part system, each part having different physical properties. The assembly is held together by a shrink-fit pressure, p_f. The outer component has a Poisson's ratio of μ_o and an elastic modulus of E_o; the inner component has properties designated μ_i and E_i. Such an assembly might, for example, be a steel shell on a brass liner.

In accordance with Equations 3 and 4 the hoop and radial stresses can be computed. These equations, however, are not sufficient to define the value of the shrink-fit pressure, consequently the deformations involved must be considered. By Equation 10 the radial deformation for the inner cylinder is

$$u_{rb} = u_i = -\frac{-p_f}{E_i}\left(\frac{a^2 + b^2}{b^2 - a^2} \pm \mu_i\right) b \quad \text{(12)}$$

Deformation for the outer cylinder is

$$u_{rb} = u_o = \frac{p_f}{E_o}\left(\frac{b^2 + c^2}{c^2 - b^2} + \mu_o\right) b \quad \text{(13)}$$

Total shrinkage allowance according to Equation 11 is the absolute sum of the individual deformations; thus, summing Equations 12 and 13,

$$\Delta = p_f\, b\left[\frac{1}{E_o}\left(\frac{b^2 + c^2}{c^2 - b^2} + \mu_o\right) + \frac{1}{E_i}\left(\frac{a^2 + b^2}{b^2 - a^2} - \mu_i\right)\right] = p_f\, b\, K \quad \text{(14)}$$

From Equation 14 the shrink-fit pressure is

$$p_f = \frac{\Delta}{bK} \quad \text{(15)}$$

2. Same Material—Hollow Shaft: For the special case where the two components are of identical materials, Equations 14 and 15 become respectively

$$\Delta = \frac{2\, p_f\, b^3\, (c^2 - a^2)}{E(c^2 - b^2)\,(b^2 - a^2)} \quad \text{(14a)}$$

$$p_f = \frac{\Delta E\ (c^2 - b^2)(b^2 - a^2)}{2\, b^3\ (c^2 - a^2)} \quad \ldots\ldots\ldots\ldots (15a)$$

3. Different Materials—Solid Shaft: As another special case, if the inner component is a solid shaft, then $a = 0$ and Equations 14 and 15 become

$$\Delta = p_f\, b \left[\frac{1}{E_o} \left(\frac{c^2 + b^2}{c^2 - b^2} + \mu_o \right) + \frac{1 - \mu_i}{E_i} \right] \quad (14b)$$

$$p_f = \frac{\Delta}{b \left[\frac{1}{E_o} \left(\frac{c^2 + b^2}{c^2 - b^2} + \mu_o \right) + \frac{1 - \mu_i}{E_i} \right]} \quad \ldots (15b)$$

4. Same Material—Solid Shaft: If the solid shaft and hub are made of the same material, Equations 14*a* and 15*a* become

$$\Delta = \frac{2\, p_f\, bc^2}{(c^2 - b^2)\, E} \quad \ldots\ldots\ldots\ldots (14c)$$

$$p_f = \frac{\Delta E\ (c^2 - b^2)}{2\, b\, c^2} \quad \ldots\ldots\ldots\ldots (15c)$$

Optimum Design: The previous analyses do not include the concept of an "optimum" design—that is, a design to give maximum performance with minimum material. In order to arrive at the best conditions the shear stress distribution in the components must be investigated.

1. Same Yield Strength—Different Materials—Hollow Shaft: For the inner shell of a two-shell system the maximum hoop stress is at the inner surface, and the maximum radial stress is at the outer boundary. Since the hoop stress is always greater than the radial stress, and both stresses are negative, the maximum shear stress is

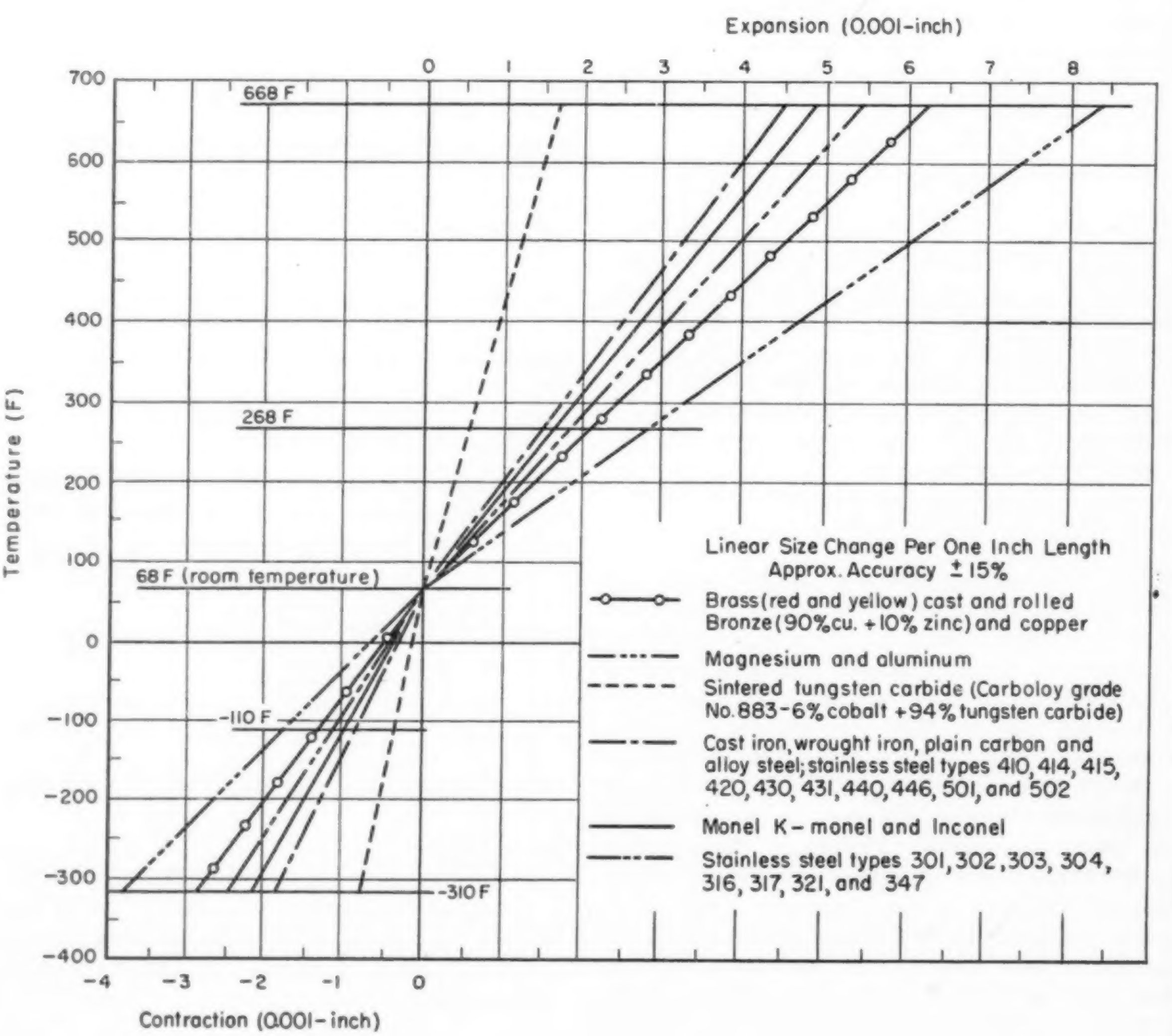
$$s_{si} = \pm \frac{\Delta}{2\,K} \left(\frac{-\,2b}{b^2 - a^2} \right) = \frac{\pm\, \Delta\, b}{K(b^2 - a^2)} \quad \ldots\ldots (16)$$

For the outer shell the maximum hoop and radial stresses occur at the interface; consequently

$$s_{so} = \pm \frac{\Delta}{2\,bK} \left(\frac{b^2 + c^2}{c^2 - b^2} \right) + \frac{\Delta}{bK} = \frac{\pm\, \Delta\, c^2}{bK(c^2 - b^2)} \quad (17)$$

For optimum design, maximum shear stress

Fig. 2—Size change due to heating or cooling



values should occur simultaneously in the inner and outer parts. In addition, this shear stress should be the shear stress at the elastic limit if full advantage of the material is to be realized. With this condition as the criterion, Equations 16 and 17 are equated in order to solve for the optimum value of the interface radius b; it is found that

$$b = \sqrt{ac} \qquad (18)$$

Substituting the value of b given by Equation 18 into Equations 16 and 17 gives

$$s_{si} = \frac{-\Delta \sqrt{ac}}{Ka(c-a)} \qquad (16a)$$

and

$$s_{so} = \frac{\Delta c}{K\sqrt{ac}\,(c-a)} \qquad (17a)$$

However, since in tension the maximum shear stress is equal to one-half the normal stress, Equations 16a and 17a can be written as

$$\frac{s_i}{2} = \frac{\Delta \sqrt{ac}}{Ka(c-a)} \qquad (16b)$$

and

$$\frac{s_o}{2} = \frac{\Delta c}{K\sqrt{ac}\,(c-a)} \qquad (17b)$$

The value of the shrinkage allowance Δ can be obtained from either Equations 16b or 17b. If the

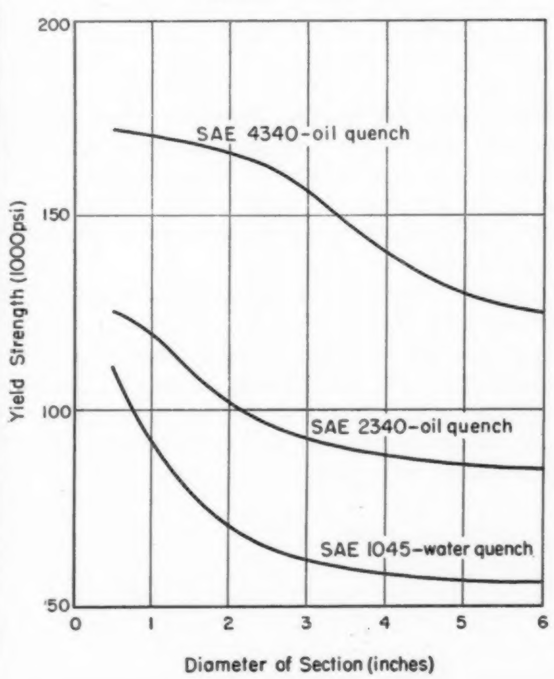


Fig. 3 — Variation in yield strength with section size of quenched steels tempered at 1000 F

yield strengths of the components are the same, then $s_i = s_o = s_y$ and the optimum shrinkage and shrink-fit pressure are determined.

The maximum shear stress theory has been utilized as a criterion for specifying the limit of elastic behavior of materials because of its convenience. It has been shown,[4,5] however, that the Distortion Energy Theory more closely fits the facts; consequently it is convenient to use the latter theory for the final results. The two theories differ only by a factor; thus where the shear theory indicates $s_y/2$ the corresponding value in terms of the more exact theory is $s_y/\sqrt{3}$. For all following calculations the shear theory will be used because of its simplicity, but the final answers will be in terms of the Distortion Energy Theory. Accordingly, for the case at hand, the optimum values are

$${}_o\Delta = s_y \sqrt{\frac{ac}{3}} \left(\frac{c-a}{c}\right)\left[\frac{1}{E_o}\left(\frac{a+c}{c-a}+\mu_o\right) + \frac{1}{E_i}\left(\frac{a+c}{c-a}-\mu_i\right)\right] \qquad (19)$$

$${}_o p_f = \frac{s_y}{\sqrt{3}}\left(\frac{c-a}{c}\right) \qquad (20)$$

2. Same Yield Strength — Same Material — Hollow Shaft: Again, if the elastic constants are the same for both parts then the optimum shrink-fit pressure is expressed by Equation 20, and the optimum shrinkage becomes

$${}_o\Delta = 2\,s_y \sqrt{\frac{ac}{3}}\left(\frac{c+a}{Ec}\right) \qquad (21)$$

Under the conditions imposed, so long as the yield strengths of the components are the same, the optimum geometry is defined as

$$\frac{c}{b} = \frac{b}{a} \qquad (22)$$

that is, the ratios of the outside to inside radii are the same for both parts. Actually, the condition expressed by Equation 22 holds for a construction of any number of shells.

3. Same Yield Strength — Same Material — Solid Shaft: In the case of an assembly having a solid shaft and constant yield strength throughout, the shear stress (Equations 16 and 17) in the shaft is always less than in the shrunk-on shell; consequently no specific optimum can be designated for this condition. It is shown later that an optimum design can be found for the case where the shaft is mechanically weaker than the hub.

4. Different Materials — Different Yield Strengths—Hollow Shaft: Another important case is that of a shrink-fit assembly in which the yield strengths are different in the component parts. For this type of construction, as for others already discussed, the elastic constants can be different for the component parts. When elastic

constants are different, the maximum shear stresses are given by Equations 16 and 17. For optimum design the maximum shear stress values should be the same in both parts, thus

$$s_{si}\left(\frac{s_{so}}{s_{si}}\right) = s_{so} \qquad (23)$$

or, using the values of yield strength given by Equation 16*b* and 17*b*, Equation 23 becomes

$$s_{si}\,\phi = s_{so} \qquad (24)$$

Substituting Equations 16 and 17 into Equation 24 gives the optimum value of the interface radius as

$$b^2 = \frac{c^2(\phi - 1) \pm c\sqrt{c^2(\phi - 1)^2 + 4\phi a^2}}{2\phi} \qquad (25)$$

Equation 25 defines the optimum interface radius for a two-shell shrink-fit construction in which the yield strength, modulus and Poisson's ratio are different in the component parts. This value of b substituted into the following equations gives new equations for defining optimum conditions, thus,

$$_o\Delta = \frac{s_o K}{b\,\phi\sqrt{3}}(b^2 - a^2) \qquad (26)$$

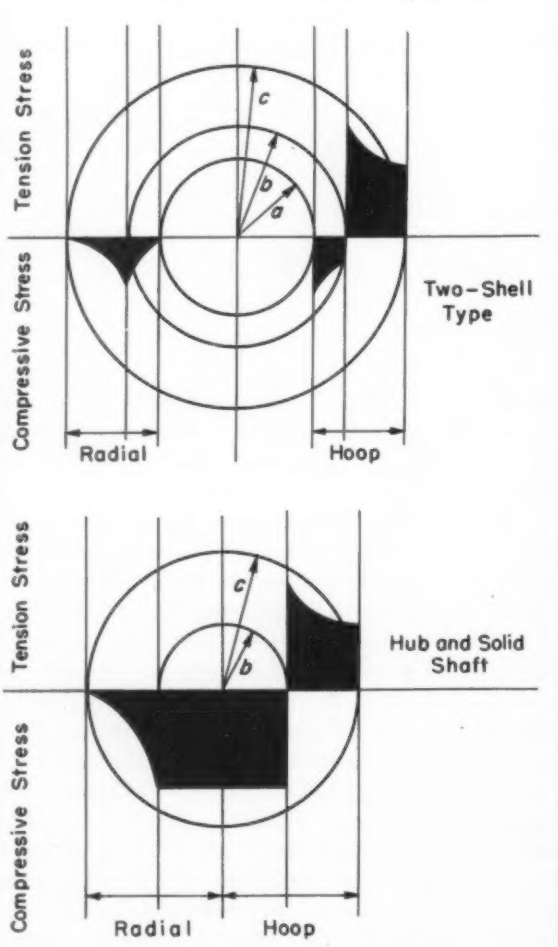


Fig. 4—Stress distribution in assemblies subjected to shrink-fit pressure

and

$$_op_f = \frac{s_o(b^2 - a^2)}{b^2\,\phi\sqrt{3}} \qquad (27)$$

5. Different Yield Strengths—Same Material—Hollow Shaft: When elastic constants are the same for all components

$$_o\Delta = \frac{2\,s_o b(c^2 - a^2)}{\phi E\sqrt{3}\,(c^2 - b^2)} \qquad (28)$$

$$_op_f = \frac{s_o}{\phi\, b^2\sqrt{3}}(b^2 - a^2) \qquad (29)$$

6. Different Yield Strengths—Different Materials—Solid Shaft: For a construction in which the inner component is a solid shaft, the value of a in Equation 26 becomes zero and

$$b = c\sqrt{\frac{\phi - 1}{\phi}} \qquad (30)$$

Equation 30 is valid only for values of $\theta > 1$ which means that if the solid shaft is made of material having a lower yield strength than that of the shrunk-on material an optimum design can be effected; consequently, for the case of a hub on a solid shaft the optimum shrinkage is obtained by substituting the value of b from Equation 30 into Equation 26, with $a = 0$. The optimum shrink-fit pressure is

$$_op_f = \frac{s_o}{\phi\sqrt{3}} \qquad (31)$$

7. Different Yield Strengths—Same Material—Solid Shaft: For designs with a solid shaft in which the yield strengths of the components are different but the elastic constants are the same, $_op_f$ is given by Equation 31 and, using Equation 30,

$$_o\Delta = \frac{2\,s_o\,c}{E\sqrt{3}}\sqrt{\frac{\phi - 1}{\phi}} \qquad (32)$$

Design for Shrink-Fit Pressure Plus Internal Pressure: Basically, the development of design formulas for pressure vessels is analogous to those previously considered; with added internal pressure, however, different conditions of maximum shear stress are involved.

1. Same Material — Same Yield Strength: Starting with the simplest case in which the yield strength, elastic modulus and Poisson's ratio are the same for both components, and using the Lame theory for cylinders it can be shown that the maximum shear stresses in the inner and outer components are, respectively,

$$s_{si} = p_i\left(\frac{c^2}{c^2 - a^2}\right) - p_f\left(\frac{b^2}{b^2 - a^2}\right) \qquad (33)$$

$$s_{so} = \frac{c^2}{b^2}\left[p_i\left(\frac{a^2}{c^2 - a^2}\right) + p_f\left(\frac{b^2}{c^2 - b^2}\right)\right] \qquad (34)$$

Optimum design requires that maximum shear stress values occur simultaneously in both parts;

consequently equating Equations 33 and 34 gives the relation

$$p_f\left(\frac{c^2}{c^2-b^2}+\frac{b^2}{b^2-a^2}\right)$$

$$= p_i\left[\frac{c^2}{c^2-a^2}\left(1-\frac{a^2}{b^2}\right)\right] \qquad (35)$$

In order to obtain the maximum shear stress in each component it is necessary to solve Equation 35 for p_f and then substitute back into Equation 33 or 34. This operation gives

$$s_s = p_i\left[\frac{b^2 c^2}{2\,b^2 c^2 - (b^4 + a^2 c^2)}\right] \qquad (36)$$

Then, using Equation 36, and the condition that by the Distortion Energy Theory of failure

$$s_s = \frac{s_y}{\sqrt{3}} \qquad (37)$$

it is found that the maximum allowable internal pressure that the compound vessel can withstand is

$${}_m p_i = \frac{2\,s_y}{\sqrt{3}}\left[1-\frac{1}{2}\left(\frac{b^2}{c^2}+\frac{a^2}{b^2}\right)\right] \qquad (38)$$

The optimum value of the interface radius b is found by differentiating Equation 38 with respect to b and equating to zero. In the optimum case

$$b = \sqrt{ac} \qquad (39)$$

Substituting the value of b given by Equation 39 into Equation 38 gives

$${}_o p_i = \frac{2\,s_y}{\sqrt{3}}\left(1-\frac{a}{c}\right) \qquad (40)$$

Equation 39 indicates that for optimum results the ratios of the outside to inside radii of both components are equal.

Knowing the optimum value of the interface radius is, however, insufficient for design; it is also required to know the optimum shrinkage involved so that proper premachining of cylinders can be accomplished. Noting Equations 15*a* and 39, the optimum shrink-fit pressure is given as

$${}_o p_f = \frac{\Delta E}{2\sqrt{ac}}\left(\frac{c-a}{c+a}\right) \qquad (41)$$

Also, from Equations 15*a*, 40 and 41

$${}_o p_f = \frac{s_y}{c\sqrt{3}}\left[\frac{(c-a)^2}{c+a}\right] = \frac{s_y}{R\sqrt{3}}\left[\frac{(R-1)^2}{R+1}\right] \qquad (42)$$

and

$${}_o \Delta = \frac{2\,s_y}{E}\sqrt{\frac{ac}{3}}\left(\frac{c-a}{c}\right) = \frac{2\,s_y\,a}{E\sqrt{3}}(R-1) \qquad (43)$$

2. Same Material—Different Yield Strengths: The foregoing is valid only if the material and properties are constant throughout the assembly. In many cases, however, the component parts have different properties. If the elastic constants are the same for both parts but the yield strengths are different, then for optimum design Equation 35 takes the form

$$\left[p_i\left(\frac{R^2}{R^2-1}\right) - p_f\left(\frac{R_1^2}{R^2-1}\right)\right]\phi$$

$$= \frac{R^2}{R_1^2}\left[p_i\left(\frac{1}{R^2-1}\right) + p_f\left(\frac{R_1^2}{R^2-R_1^2}\right)\right] \qquad (44)$$

With Equation 44 defining the value of p_f, and by using this value of p_f in Equations 33 or 34 and 37, the maximum internal pressure which can be sustained by the compound cylinder without elastic failure is

$${}_m p_i = \frac{s_i}{\sqrt{3}}\left(1+\phi-\frac{1}{R_1^2}-\frac{R_1^2}{R^2}\phi\right) \qquad (45)$$

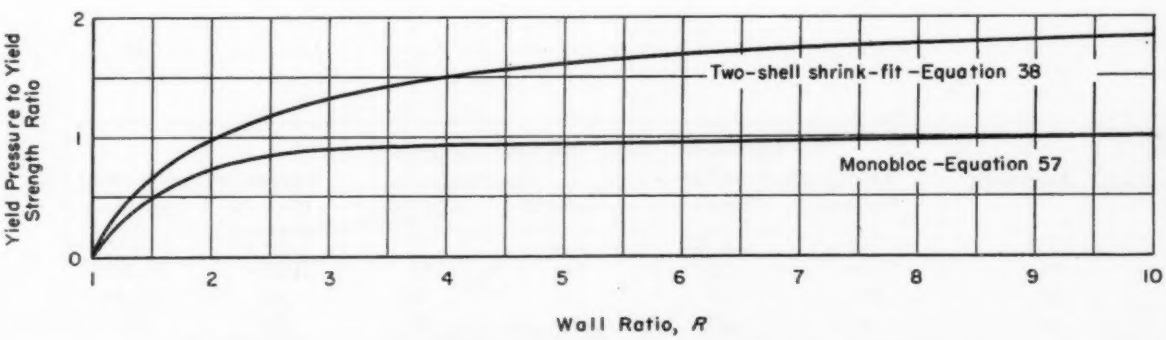
Differentiating Equation 45 with respect to b and equating to zero gives the optimum interface radius as

$$b^2 = \frac{ac}{\sqrt{\phi}} \qquad (46)$$

or

$$R_1^2 = \frac{R}{\sqrt{\phi}} \qquad (47)$$

Fig. 5—Elastic-breakdown characteristics of heavy-wall cylinders



Substituting Equation 47 into Equation 45 gives the optimum internal pressure, or maximum internal pressure based on optimum conditions to just avoid yielding of the assembly; thus

$$_op_i = \frac{2\,s_i}{\sqrt{3}}\left(\frac{1+\phi}{2} - \frac{\sqrt{\phi}}{R}\right) \quad \text{(48)}$$

Using Equations 48 and 15 with proper substitutions to solve for K the optimum shrink-fit pressure and shrinkage become

$$_op_f = \frac{s_i}{R\sqrt{3}\,(R^2-1)}\,(R_1{}^2\,\phi - 1)^2\,(R - \sqrt{\phi}) \quad \text{(49)}$$

and

$$_o\Delta = \frac{2\,s_ib}{ER\sqrt{3}}\left[\frac{R\sqrt{\phi}-\phi}{\sqrt{\phi}\,(R^2+1) - R\,(\phi+1)}\right] \times$$

$$[1 + R(R\phi - 2\sqrt{\phi})] \quad \text{(50)}$$

3. Different Materials—Same Yield Strength: When yield strengths are the same in the two parts but the elastic constants differ, the equations used are

$$_mp_i = \frac{2\,s_y}{\sqrt{3}}\left(1 - \frac{a}{c}\right) = \frac{2\,s_y}{\sqrt{3}}\left(\frac{R-1}{R}\right) \quad \text{(51)}$$

$$b = \sqrt{ac} \quad \text{(52)}$$

$$_op_f = \frac{s_y}{c\sqrt{3}}\left[\frac{(c-a)^2}{c+a}\right] = \frac{s_y}{R\sqrt{3}}\left[\frac{(R-1)^2}{R+1}\right] \quad \text{(42)}$$

$$_o\Delta = \frac{s_y}{c}\sqrt{\frac{ac}{3}}\left[\frac{(c-a)^2}{c+a}\right] \times$$

$$\left[\frac{1}{E_o}\left(\frac{a+c}{c-a} + \mu_o\right) + \right.$$

$$\left.\frac{1}{E_i}\left(\frac{a+c}{c-a} - \mu_i\right)\right] \quad \text{(53)}$$

4. Different Materials — Different Yield Strengths: Similarly, for the case where both materials and properties are different,

$$_mp_i = \frac{2\,s_i}{\sqrt{3}}\left(\frac{1+\phi}{2} - \frac{\sqrt{\phi}}{R}\right) \quad \text{(54)}$$

$$b^2 = \frac{ac}{\sqrt{\phi}} \text{ or } R_1{}^2 = \frac{R}{\sqrt{\phi}} \quad \text{(55)}$$

$$_op_f = \frac{s_i}{R\sqrt{3}\,(R^2-1)}\,(R_1{}^2\,\phi - 1)^2(R - \sqrt{\phi}) \quad \text{(49)}$$

$$_o\Delta = \frac{s_ibk}{R\sqrt{3}\,(R^2-1)}\,(R_1{}^2\phi - 1)^2(R - \sqrt{\phi}) \quad \text{(56)}$$

Summary: Because of the number of equations involved, they have been summarized in Table 2.

The practical use of the foregoing theory is demonstrated by tests conducted at the Du Pont Engineering Research Laboratory. Cylinders were fabricated having ratios of outside to bore radius of 2.75 and 4.00. Two cylinders were of monobloc type construction and the other two were of the two-shell shrink-fit type. Mechanical properties and pertinent geometrical data concerning these cylinders are presented in Table 3. In testing the assemblies the ends were sealed,[4] and curves of external surface hoop strain versus internal pressure were plotted. The initial portion of the pressure-strain curve is linear in accordance with Lame theory, but at the beginning of yielding of the cylinder, deviation from linearity occurs; this pressure is called the "elastic-breakdown pressure." This value can be predicted with fair accuracy both for monobloc and shrink-fit type cylinders as shown in Table 4. For the monobloc cylinders

$$p_i = \frac{s_y}{\sqrt{3}}\left(\frac{R_{ob}{}^2 - 1}{R_{ob}{}^2}\right) \quad \text{(57)}$$

Table 2—Summary of Design Equations

Shrink-Fit Pressure Only

Assembly	Shrink-Fit Pressure General	Shrink-Fit Pressure Optimum	Shrinkage General	Shrinkage Optimum
Hollow shaft				
(1) $E_i = E_o,\ \mu_i = \mu_o,\ s_i = s_o$	15a	20	14a	21
(2) $E_i = E_o,\ \mu_i = \mu_o,\ s_i \neq s_o$	15a	29	14a	28
(3) $E_i \neq E_o,\ \mu_i \neq \mu_o,\ s_i \neq s_o$	15	27	14	26
(4) $E_i \neq E_o,\ \mu_i \neq \mu_o,\ s_i = s_o$	15	20	14	19
Solid shaft				
(1)	15c	..	14c	..
(2)	15c	31	14c	32
(3)	15b	31	14b	26, 30
(4)	15b	..	14b	..

Shrink-Fit plus Internal Pressure

Assembly	Shrink-Fit Pressure General	Shrink-Fit Pressure Optimum	Shrinkage General	Shrinkage Optimum	Internal Pressure, Max General	Internal Pressure, Max Optimum
(1)	15a	41, 42	14a	43	38	40
(2)	15a	49	14a	50	45	48, 54
(3)	15	49	14	56	45	48, 54
(4)	15	42	14	53	38	51

For the shrink-fit cylinders, as shown in TABLE 3, the shells and liners had different mechanical properties; consequently the optimum shrink-fit interference was calculated using Equations 46 and 47. Having the required interference, the temperature differential required in order to expand the shell by the amount of the interference was calculated using Equation 1. The pressure at elastic failure can be calculated from Equation 48.

Through utilization of the previous concepts, design characteristics of the two types of construction can easily be compared. Characteristics of two-shell shrink-fit vessels under internal pressure when the materials and properties are the same for both parts will be used as one case; monobloc construction as the other. The elastic response of both monobloc and two-shell vessels under internal pressure is shown in *Fig.* 5. The ordinate can be considered as representing the elastic-breakdown pressure, while the abscissa represents the ratio of outside to bore radius (wall ratio). The curve for the monobloc cylinder shows that little, if any, gain can be expected in elastic-breakdown pressure by increasing the wall ratio beyond about 3.50 or 4.00. By shrink-fitting some advantage is gained, particularly in the low wall ratio range; however, the advantage tapers off at about a wall ratio of 6 or 7. The shrink-fit construction is always better than the monobloc, however, and becomes twice as good at infinite wall ratio. In order to determine the equivalence of the two types of construction, *Fig.* 6 shows the wall ratios required for both types of vessels to give the same elastic-breakdown pressure. The latter curve shows, for example, that a monobloc vessel of wall ratio 6.00 can be replaced by a two-shell shrink-fit type of wall ratio 1.95. Similar curves can be constructed for the case where either the materials are different or the mechanical properties of the two parts are different. For example, the elastic-breakdown pressure can be increased over one-third by increasing the strength of the outside shell from $0.5s_i$ to $1.25s_i$.

Table 3—Properties and Geometry of Test Cylinders*

Property or Dimension	Monobloc 1	Monobloc 2	Shrink-fit 3	Shrink-fit 4
ID (in.)	1.501	1.377	1.498	1.374
OD (in.)	4.128	5.502	4.128	5.502
Interface diam (in.)	....	....	2.50	2.75
Overall wall ratio (in.)	2.75	4.00	2.756	4.01
Liner wall ratio (in.)	....	....	1.67	2.00
Shrinkage (0.001-in./in.)	....	....	2.84	3.49
Yield strength, shell (1000 psi)	125	113	125	125
Yield strength, liner (1000 psi)	....	....	103	103

*Cr-Ni-Mo-V steel, quenched and tempered at 1100 F.

Miscellaneous Shrink-Fit Problems

The important miscellaneous shrink-fit problems such as effect of centrifugal forces, axial and torsional holding ability have been previously treated;[1] however, for completeness here a brief review will be made of some of the salient features.

Effect of Centrifugal Forces: Shrink-fit assemblies are often used in applications where high speeds are encountered; consequently it is necessary in design to take this effect into account in shrink-fit calculations. The effect of rotation is to develop high centrifugal forces which tend to

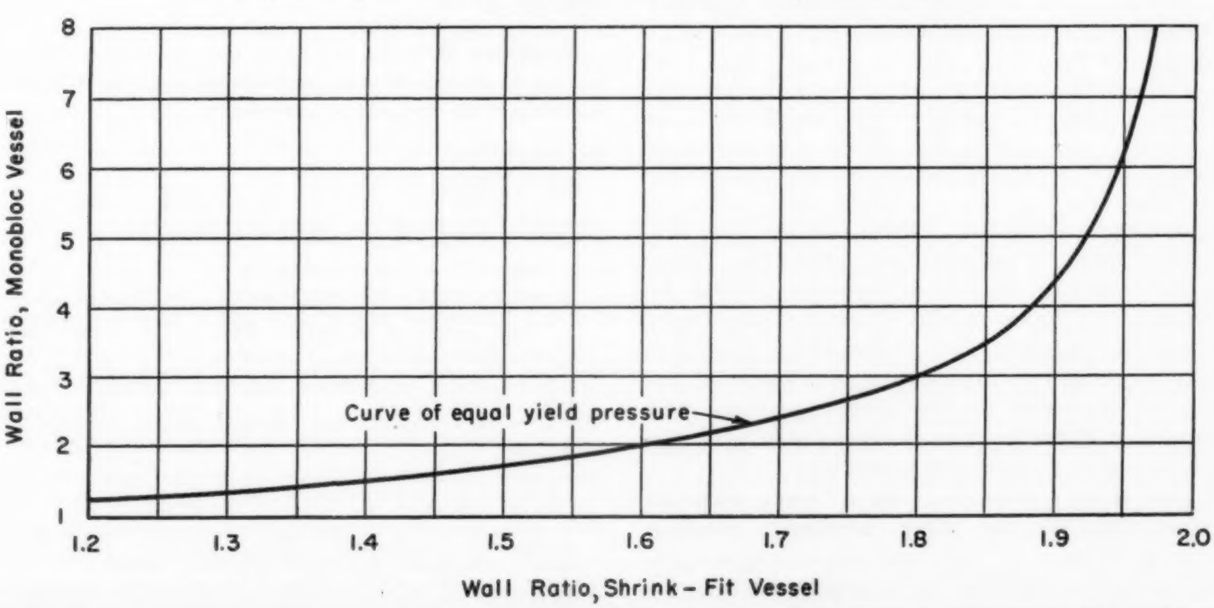


Fig. 6—Equivalence of monobloc and shrink-fit pressure vessels

change the shrinkage between the two parts and thus influence the interface pressure.

1. Same Material—Hollow Shaft: The compound cylinder can be considered as composed of two parts, an inner component which develops a radial deformation at b of u_i and an outer component which develops a radial deformation at b of u_o. From the theory of elasticity[5] these deformations are

$$u_i = \frac{N}{8}\left[-b^3 + b\,(b^2 + a^2)\left(\frac{3+\mu}{1+\mu}\right) + a^2 b\left(\frac{3+\mu}{1-\mu}\right)\right] \quad (58)$$

$$u_o = \frac{Nb}{4\,(1-\mu^2)}\,[b^2\,(1-\mu) + c^2\,(3+\mu)] \quad (59)$$

where

$$N = (1-\mu^2)\,\frac{\gamma\omega^2}{g\,E} \quad (60)$$

The change in shrinkage because of centrifugal action is

$$\Delta' = \mu_o - \mu_i \quad (61)$$

which gives

$$\Delta' = \frac{b\gamma V^2}{4\,g\,E}\,(3+\mu)\left(1 - \frac{a^2}{c^2}\right) \quad (62)$$

2. Different Materials—Hollow Shaft: When the elastic constants are different in the two parts,

$$\Delta' = \frac{V^2 b}{4\,g}\left\{\frac{\gamma_o}{E_o}\left[(3+\mu_o) + (1-\mu_o)\frac{b^2}{c^2}\right] - \frac{\gamma_i}{E_i}\left[(3+\mu_i)\frac{a^2}{c^2} + (1-\mu_i)\frac{b^2}{c^2}\right]\right\} \quad (63)$$

3. Same Material—Solid Shaft: For the special case of a solid shaft, $a = 0$ in the above equations and

$$\Delta' = \frac{V^2\,b\,\gamma\,(3+\mu)}{4\,g\,E} \quad (64)$$

4. Different Materials—Solid Shaft: For different materials in the hub and shaft,

$$\Delta' = \frac{V^2 b}{4\,g}\left\{\frac{\gamma_o}{E_o}\left[(3+\mu_o) + (1-\mu_o)\frac{b^2}{c^2}\right] - \frac{\gamma_i}{E_i}\,(1-\mu_i)\,\frac{b^2}{c^2}\right\} \quad (65)$$

From the standpoint of optimum design, centrifugal action has an influence on purely static calculation. Calculations should therefore be made to determine if optimum conditions can be maintained at operating speeds. This is done by noting the change in shrinkage that occurs as a result of rotation, as expressed by the above equations, and determining if the change is compatable with the development of an optimum shrinkage.

If the inner component has a lower modulus of elasticity than the outer component, rotation increases pressure at the interface; it may be possible initially to prepare the assembly with a corresponding decrease in shrinkage so that, at operating speed, optimum conditions will be realized.

Table 4—Comparison of Actual and Calculated Results for Test Cylinders

Construction	Elastic Breakdown Pressure Observed (1000 psi)	Calculated (1000 psi)	Deviation (percent)
Monobloc 1	65	63	−4
Monobloc 2	65	71	+9
Shrink-fit 3	76	82	8
Shrink-fit 4	85	96	13

If the outer component has a lower or the same modulus of elasticity, the two parts tend to separate during rotation, and the interface pressure decreases. In this case, optimum design at operating speed may not be possible since the initial increase in shrinkage, to compensate for the decrease in shrinkage at operating speed, may result in plastic deformation of the assembly before operation with accompanying redistribution of stress and departure from optimum conditions.

A shrink-fit assembly made up of two hollow disks and subjected to centrifugal action is shown in *Fig. 1d*. In order to determine the conditions under which the two components would separate, the deformations involved must be considered. The radial deformations at the interface due to centrifugal action are given by Equations 58 and 59. The same deformations due to the shrink-fit pressure are given by Equations 12 and 13. The condition for loosening is thus

$$\text{Equations } 58 + 12 = \text{Equations } 59 + 13 \quad (66)$$

Axial Holding Ability: The axial force which a shrink-fit assembly can withstand without separation of the parts is

$$F = 2\pi L b p_f f \quad (67)$$

For determining various degrees of axial holding ability (with the coefficient of friction known) the value of the shrink-fit pressure in Equation 67 can be replaced by its value as given by previous Equations.

Torsional Holding Ability: The torque which a shrink-fit assembly can withstand without relative movement of the parts is given by

$$T = 2\pi L b^2 p_f f' \quad (68)$$

The coefficient of friction in torsion must usually be found by experimental determination.[1, 6]

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Stimulating Invention by ENGINEERS

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Patents Can . . .

. . . protect development and promotional expenditures by barring competitors from production of the same product for a certain length of time.

. . . convince investors that your company has other than the usual material assets.

. . . build good will and prestige with customers, government and the general public.

. . . convince employees of your company's progressive attitude.

. . . be sold, used to obtain cross licensing agreements or produce income from royalties.

. . . stimulate employees to invent and give them the self-satisfaction of having their inventions recognized and patented.

CONTINUED success of many companies requires invention by their engineers, whether the ultimate end is cost reduction, increased reliability or entirely new products. Engineers are, of course, paid a wage or salary for just such invention, but may not perform at full efficiency if their efforts do not receive adequate recognition. Often patents will be found to be the only form of recognition available to certain employees.

Positive Policy: When a company has an affirmative patent policy which urges its employees to turn in their ideas for patent purposes, its engineers have an understanding of the patent system and of its requirements. Employees know that invention is a part of their duties and that the company realizes that there are ways of improving upon the present product and way of doing things. A definite patent policy gives employees a feeling that they are participating in the growth of their company and the industry of which it is a part.

A positive patent policy can also provide valuable information to speed invention or design. More than 2,600,000 issued U. S. patents form an immense storehouse of technical knowledge. Determining how the other fellow did it and starting where he left off releases time for more useful endeavors.

A patent search which uncovers relevant expired patents will indicate the availability of disclosures which may be used as they are or used as a basis for new improvements. Discovery of relevant adversely owned unexpired patents may well stimulate original design because of the necessity of designing around such patents.

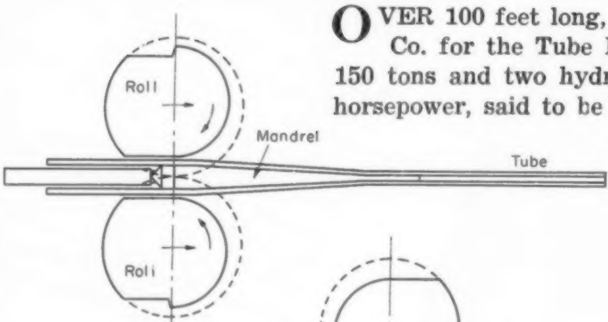
Design may be stimulated and speeded by using the patent system to uncover what others have done and to protect whatever the company is currently doing. It is then not necessary to charge a development cost to one particular machine.

Negative Policy: A company which says, "Why bother with the patent situation? We'll go ahead anyway," says in effect, that it does not believe in the patent system, or the capabilities of its employees, or both. Such indifference on the part of the employer may engender indifference in the minds of its employees, or alternatively may lead to company thinking which is out of step with the thinking of its employees. The company's engineers, at least, will inherently know or soon find out that excellent inventions can be commercially wrecked if they are improperly handled and that even a small advance in a field may be made surprisingly valuable if properly handled.

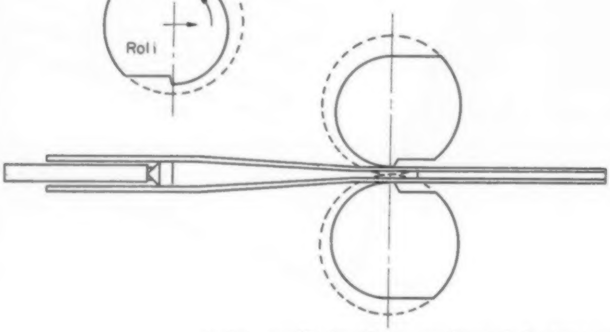
The company without the detached objectivity of a patent policy (or a really workable suggestion system) does not stimulate the free flow of ideas from its employees. Anyone who has progressed up the rungs of a company management ladder can testify that the routine channels of organizational communication are not usually adequate or suited to such purposes. However, consultation with the company's own patent lawyer regarding a new idea will not only be accepted but appreciated.

Summary: The patent system may be used to stimulate invention while serving its many other purposes commercial, financial, psychological and technical. This may have a salutory effect on the scientific worker because it gives meaning to the real work of invention. Additionally, engineers who consult a patent attorney at various stages of a development will probably find that the attorney can often help, stimulate and guide the lines that the particular development should take.

Giant Machine Cold Reduces 18-inch Tubing



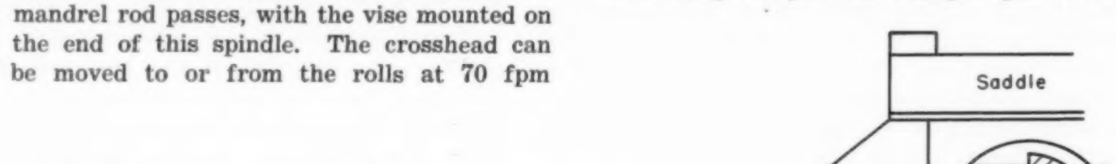
OVER 100 feet long, a new tube reducer being built by E. W. Bliss Co. for the Tube Reducing Corp. has a moving saddle weighing 150 tons and two hydraulic pumps capable of delivering up to 4000 horsepower, said to be the largest of their type ever produced. The machine can cold reduce tubing from 10 to 18 inches in diameter to produce finished tubing 9 to 16 inches OD.



Basis for the machine is the method of cold-reducing tubing shown. Fundamentally, the process consists of passing rolls or dies with tapered grooves over the tubing, which is supported by a mandrel. Each groove is slightly larger than the ingoing tube at one end and equal to the finished-tube size at the other. The tube wall is thus reduced between the mandrel and the rolls in a series of intermittent longitudinal passes. To prevent ovality, since each pass compresses the tube elliptically rather than as a true circle, the tube is rotated about 60 degrees after each pass and fed in slightly.

Four principal sections make up the whole machine: (1) the tube feeding equipment, (2) mandrel handling equipment, (3) the roll housing section—saddle and side frames, and (4) drive for the saddle.

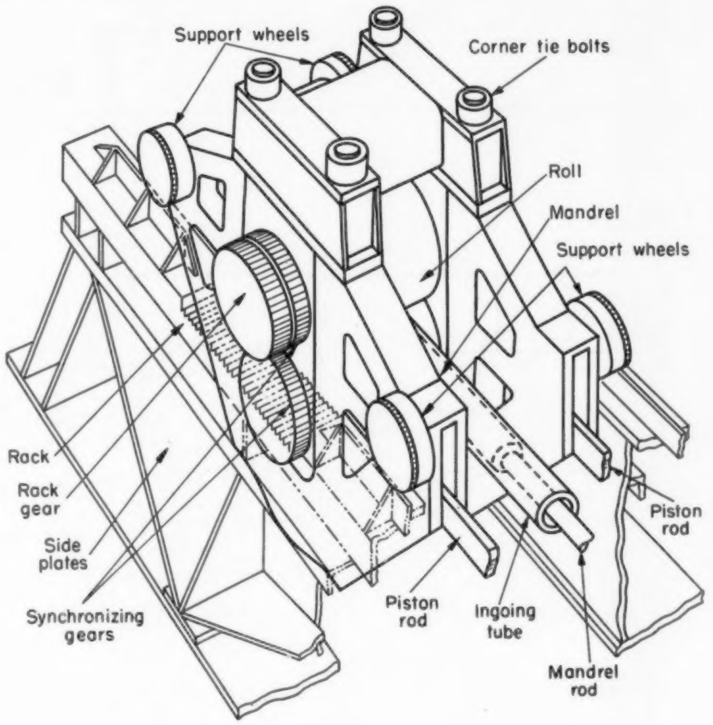
To obtain controlled tube feeding, since the tube is not moved through the pass by the action of the rolls, one end of the tube is secured in a vise on a crosshead. The crosshead, free to move on ways approaching the rolls, contains a hollow spindle through which the mandrel rod passes, with the vise mounted on the end of this spindle. The crosshead can be moved to or from the rolls at 70 fpm for preliminary operations, or can be advanced in increments of 0.1 to 0.5-inch with the advance timed by cams. A pair of nonrotating feed screws advance the crosshead, driven by feed screw nuts at the rear of the crosshead ways. These nuts are driven by a single 200 horsepower piston-displacement hydraulic motor operating at 2000 psi, and equipped with a decelerating feature to provide for high inertia. A large accumulator containing compressed nitrogen gas at 2000



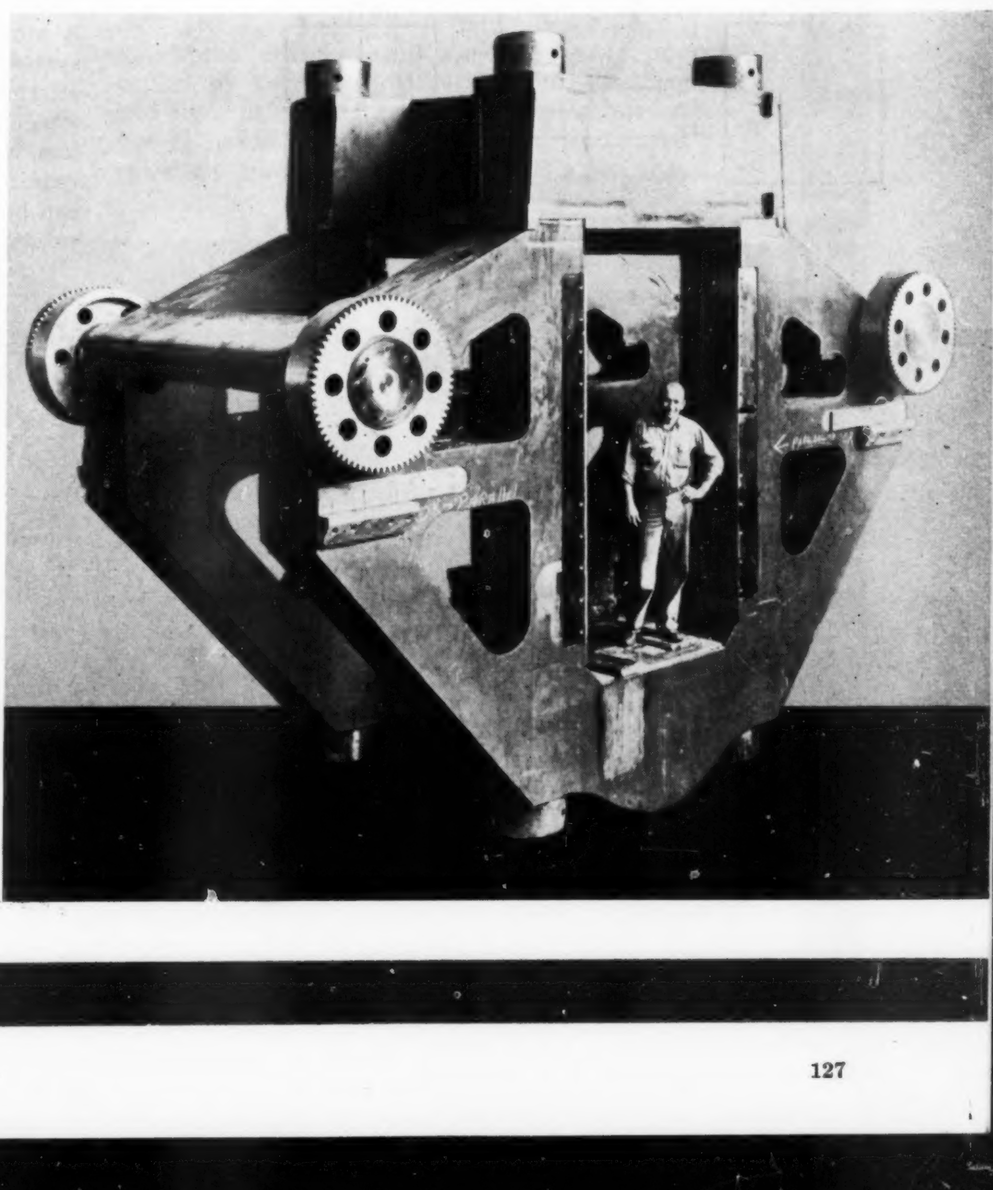
to 3000 psi provides uniform pressure.

The tube is turned at three points—by the crosshead vise, by a vise on the tube at the outlet, and by the mandrel. Uniform turning is accomplished by 325-horsepower hydraulic motors, one arranged to turn both the mandrel and a hexagonal shaft running the length of the crosshead travel which rotates the crosshead vise, and the other to turn the outlet vise.

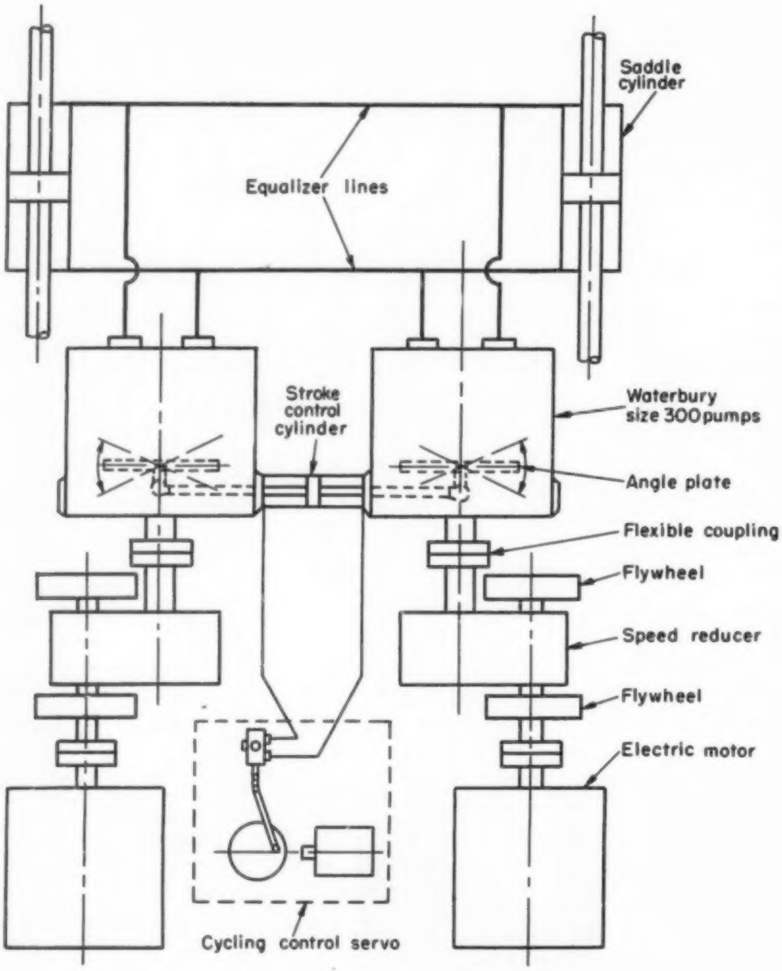
Since the mandrel bar must be withdrawn and reinserted through each new tube, the rear end of the bar is mounted on a slide which can be pulled back by a hydraulically driven chain at 75 fpm. It is locked in place for accurate positioning by a pair of hydraulically operated slides which push it forward against an adjustable stop.



Weighing 150 tons, the saddle resembles a 50-inch mill roll housing equipped with four 36-inch wheels which suspend it from rails 7 inches above the pass line. Maximum travel is about 73 inches, permitting the 50-inch rolls to rotate slightly less than 180 degrees. Corner posts, 10 inches in diameter, contain Calrod heating units so that they can be expanded lengthwise by heating; after the nuts are tightened, cooling and consequent shrinkage creates a large compression preload on the saddle frame to resist separating forces.



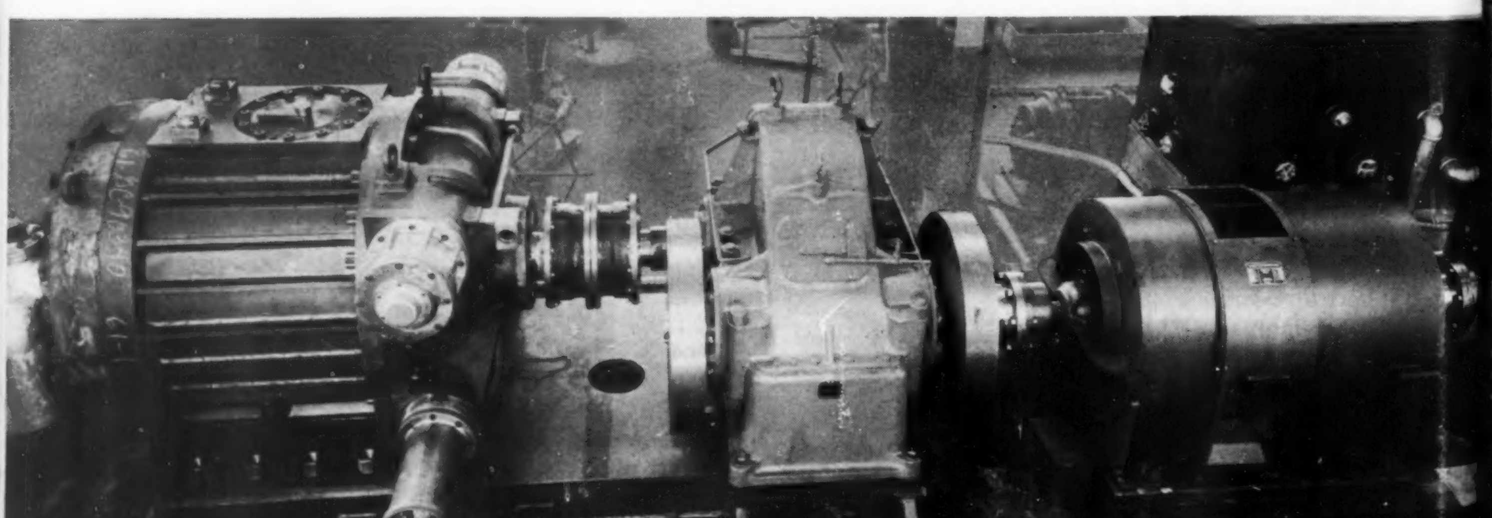
Top roll position is fixed, and the bottom roll is adjusted vertically toward the pass line by hydraulic cylinders acting like jacks under the roll bearing chucks. Since the dies run face on face, the cylinders can be preloaded to maintain position up to a specific maximum separating load. SKF double-roller spherical bearings, 48 inches OD, weighing 5150 pounds each, and having a load rating of 3.2 million pounds, are used. Top roll is driven by gears engaging stationary racks, with synchronizing gears running the bottom roll.



Requirements for the reciprocating saddle drive included variable length stroke, adjustable position of shortened stroke, progressive shortening of stroke at a controlled rate, and speed variable as a function of stroke length for maximum production. To add to the problem was the tremendous load on the drive system necessary to accelerate and decelerate the 150-ton saddle.

Saddle reciprocation is accomplished by a pair of 13-inch hydraulic cylinders operating at a working pressure of 2200 psi, and requiring 5400 gpm of oil. Each cylinder is driven by a pump developed by Waterbury Tool Co., with equalizing lines maintaining co-ordinated movement. Reversal of flow is accomplished without valves or accumulators by controlling flow from the pump. A servomechanism controls length and frequency of the stroke.

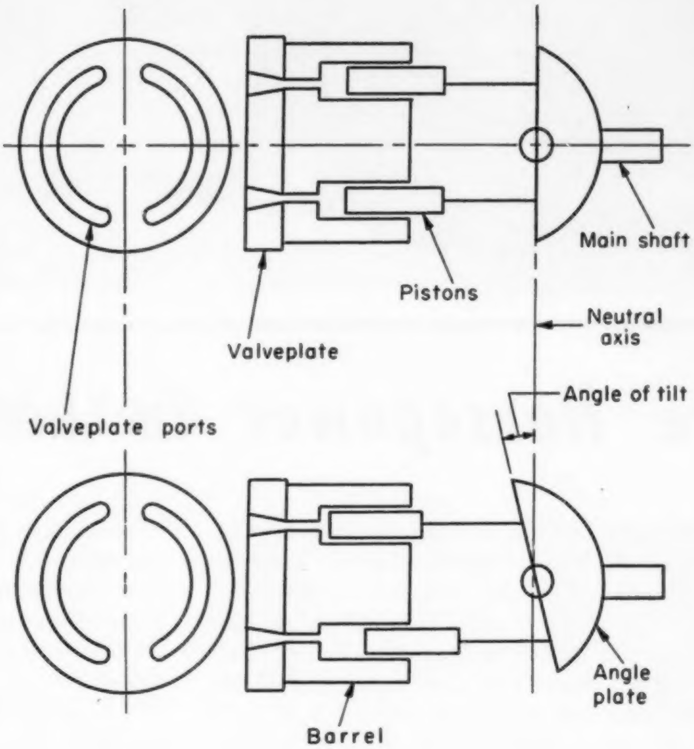
Pumps are driven through a 4.83 to 1 speed reducer by individual 700-horsepower synchronous motors. Flywheels on the high-speed shafts of the speed reducers, and the saddle itself, constantly store and supply energy back to the system, with the stored energy passing alternately from the saddle to the flywheel and back. Without such a system, a 7500-horsepower motor might be required. In case of a jam, the saddle can be cut loose, by means of overload safety valves, from the stored energy of the drive system.

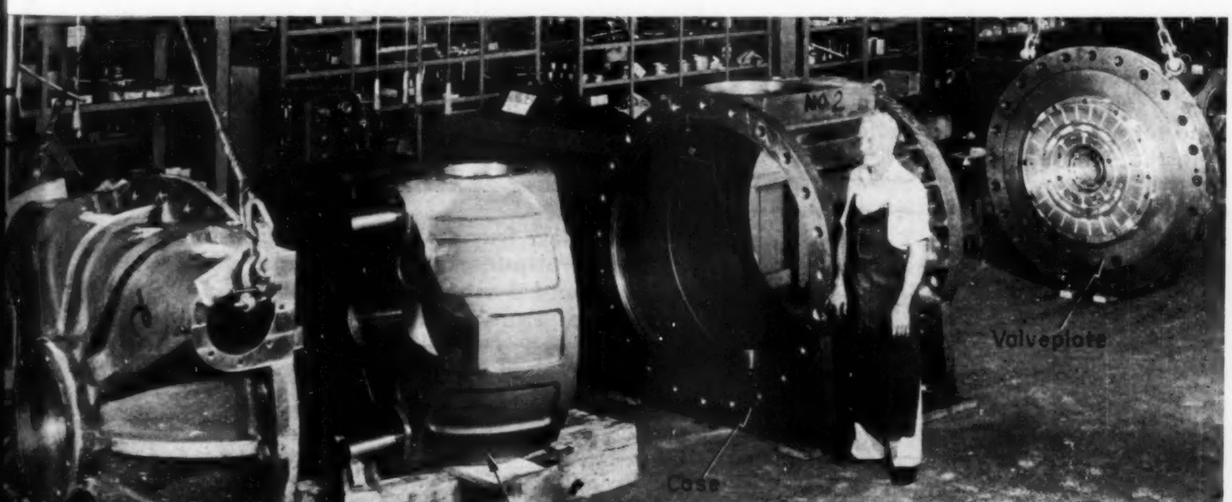


Each of the two pumps can deliver up to 2300 gpm in either direction of flow at up to 3000 psi and will operate as either a pump or hydraulic motor. Of the axial-piston, variable-displacement, reversible angle-plate type, the pumps each have 13 cylinders using 4-inch diameter pistons. The pump moves from zero to full stroke during the first half of the saddle movement in one direction, and from full stroke to zero in the second half, during which time the pump runs as a motor and acts as a brake.

Rate and direction of oil flow depend on angle of the wobble plates. The wobble plate crosses the neutral axis at the beginning of the stroke, and full angle of tilt is reached at the middle of the stroke, giving maximum flow. As the wobble plate moves back toward the neutral position, inertia of the saddle causes back pressure, which causes the pump to act as a motor and slows down the saddle.

A servomechanism controls wobble-plate motion, with the phase shift between the saddle and pump providing the controlling force. Stroke length and frequency are controlled by the same mechanism through the speed and angle at which the wobble plates are moved relative to saddle motion. The control mechanism provides any speed from zero to 45 cycles per minute.





Photo, courtesy, General Electric Co.

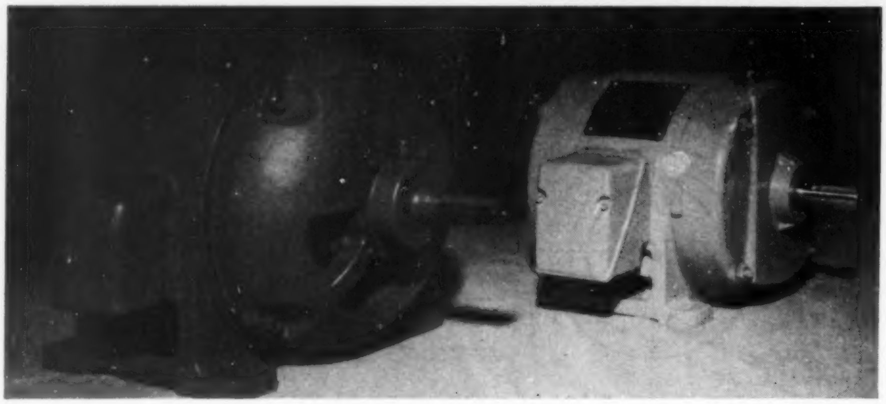


Fig. 1—Both volume and weight reduction result from the new NEMA standards. Motors shown here are old and new 2-horsepower models

More Horsepower in Less Space With

FIRST electric motors meeting the new NEMA (National Electrical Manufacturers Association) standards become available this month (see p. 204, Nov. and p. 262, Dec., MACHINE DESIGN). These new motors will produce more horsepower in a given frame size than previous motors, *Fig.* 1. In some instances size reduction will be more than 50 per cent by volume with weight reduction of as much as 40 per cent. Despite these large reductions in both volume and weight, the motors will satisfy the same performance standards of temperature rise, torque and starting current limitation.

What advances in motor building techniques permit such drastic reductions in size and weight? Superior synthetic insulating materials are a large contributor. Superior electric steels and lamination treatments permit much higher field strengths for a given weight of steel. Improved die casting techniques produce superior aluminum squirrel cage motor rotors. Clever engineering applied to design of housings produces housings with greater strength per pound of metal. All these combine to produce smaller, TABLE 1, and lighter motors.

Although the properties of copper conductors that are the most important part of the motor remain the same, superior insulating materials produce the effect of an improvement. Their lack of bulk allows more actual copper to be used by eliminating much of the space previously taken up by bulky insulation. Cooling is also improved by the new insulation materials because heat transfer characteristics are better.

Advantages to motor users and designers resulting from the rerating program include size and

Table 1—Comparison of new and old frame sizes of polyphase, squirrel cage, general-purpose open-type and totally enclosed fan-cooled motors in the 1 to 30-hp range

Hp	Open-Type		TEFC	
	New	Old	New	Old
1	182	203	182	203
1½	184	204	184	204
2	184	224	184	224
3	213	225	213	225
5	215	254	215	254
7½	254U	284	254U	284
10	256U	324	256U	324
15	284U	326	284U	326
20	286U	364	286U	364
25	324U	364	326U	365
30	326U	365	...	404

Note: "U" following frame number pertains to shaft size.

Table 2—Suggested availability dates for motors meeting latest NEMA standards

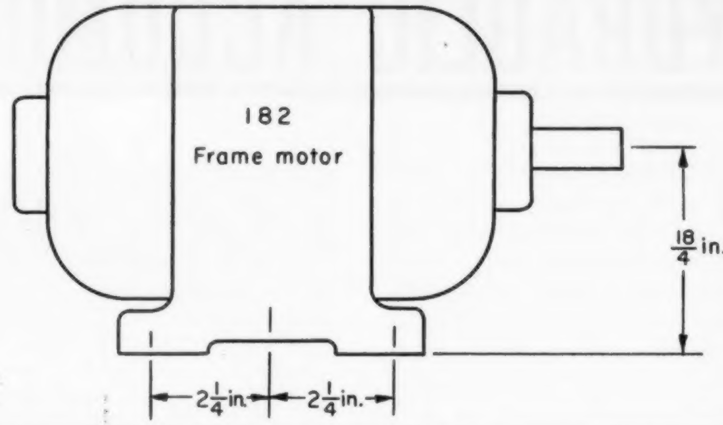
Frame No.	Availability Date
182, 184	Jan. 1, 1954
213, 215	June 1, 1954
254U, 256U	Nov. 1, 1953
284U, 286U	April 1, 1955
324U, 326U	Sept. 1, 1955

Table 3—New frame sizes for polyphase, squirrel cage, general-purpose horizontal and vertical Design A, B and C motors, open-type rated to 40 C temperature rise for 550 volts or less, 60 cycles

Hp	Speed (rpm)					
	3600	1800	1200	900	720	600
½	...	...	...	182*	...	...
¾	...	...	182*	184*	...	...
1	...	182*	184*	213*	213*	215*
1½	182*	184*	184*	213*	215*	254U*
2	184*	184*	213*	215*	254U*	256U*
3	184*	213*	215*	254U	256U*	286U*
5	213*	215	254U	256U	286U*	324U*
7½	215*	254U	256U	284U	324U	326U*
10	254U*	256U	284U	286U*	326U*	...
15	256U*	284U	324U	326U	...	...
20	284U	286U	326U	...	...	...
25	286*	324U	...	...	...	...
30	324S*	326U	...	...	...	...
40	326S*	...	...	...	...	...

*Applies to Design A and B motors only.
Note: U or S following frame number pertains to shaft size or extension.

Fig. 2—Frame size numbers relate to shaft height and mounting hole location



New Electric Motors

weight reduction, of course. Additionally, motors made in conformance with the new standards should be superior to previous motors in other respects. As an example of this, one motor manufacturer claims higher efficiency, higher full-load speed ratings and less motor noise for the new models. Another quite welcome benefit forecast by a large motor manufacturer is the reduction of the tendency of electric motor prices to follow the present inflationary curve, because of the elimination of superfluous, nonproductive masses of metal.

Table 4—New frame sizes for single-phase, 115 and 230-v 60-cycle ac, general-purpose, horizontal and vertical motors, open-type rated 40 C temperature rise

Hp	Speed (rpm)		
	3600	1800	1200
¾	...	...	184
1	...	184	213
1½	184	184	215
2	184	213	...
3	213	215	...
5	215	254U	...
7½	254U	256U	...

Table 5—New frame sizes for polyphase, squirrel cage, horizontal and vertical Design A, B and C motors, totally enclosed, fan-cooled type rated 55 C temperature rise for 550 v or less, 60 cycles

Hp	Speed (rpm)					
	3600	1800	1200	900	720	600
½	...	...	...	182*	...	...
¾	...	...	192*	184*	...	...
1	...	182*	184*	213*	213*	215*
1½	182*	184*	184*	213*	215*	254U*
2	184*	184*	213*	215*	254U*	256U*
3	184*	213*	215	254U	256U*	286U*
5	213*	215	254U	256U	286U*	324U*
7½	215*	254U	256U	284U	324U*	326U*
10	254U*	256U	284U	286U	326U*	...
15	256U*	284U	324U	326U*	...	...
20	286U*	286U	326U*	...	...	...
25	326S*	326U	...	...	...	...

*Applies to Design A and B motors only.
Note: U or S following frame numbers pertains to shaft size or extension.

Although a complete rerating program has been undertaken for all popular ac motors from 1 to 40 horsepower, TABLES 3, 4, 5 and 6, only those in frame sizes 182 and 184 will be available this month, because of the substantial amount of engineering and retooling necessary for the production of the new designs. A suggested schedule, TABLE 2, for the production of the new types calls for one larger frame diameter at five month intervals starting in January, 1954. This schedule, if followed, would make the largest sizes available in September, 1955.

NEMA FRAME SIZE NUMBERS: The first two digits of the NEMA frame size number are approximately equal to four times the shaft height, or distance from the mounting surface to the centerline of the shaft of a horizontal motor. The last digit of the NEMA frame size number is an indication of the distance from the center of the frame to the mounting holes measured parallel to the shaft. A 182 frame motor, for example, *Fig.* 2, has a shaft height of 18/4 or 4½ inches, or slightly less, so the exact height if necessary, can be obtained by shimming. The distance from frame centerline to mounting hole center for this frame is indicated by the third digit, 2, and is 2¼ in., according to the NEMA standard dimension chart.

Table 6—New frame sizes for polyphase, wound-rotor general-purpose horizontal and vertical motors, open-type rated 40 C temperature rise, 60 cycles

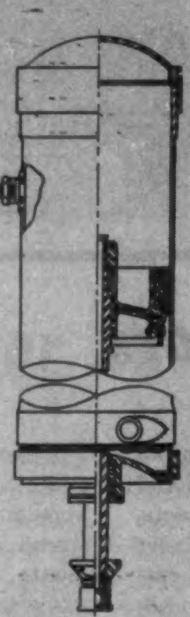
Hp	Speed (rpm)		
	1800	1200	900
¾	...	...	213
1	...	...	215
1½	...	213	254U
2	213	215	254U
3	215	254U	256U
5	254U	256U	284U
7½	256U	284U	324U
10	284U	324U	326U
15	324U	326U	...
20	362U	...	...

Note: *U* following frame No. pertains to shaft size.

HYDRAULIC ACCUMULATORS

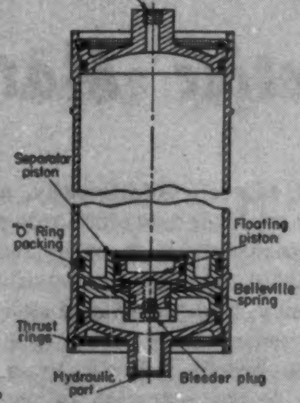
MODERN developments in aircraft, and also in self-contained machines generally, firmly established the basic requirement for some form of dynamic hydraulic power reservoir. Earliest work in this field developed the inflexible separator hydropneumatic accumulators and finally led to the latest flexible separator types. These offer maximum power output and reliability with minimum weight and size, features of increasing importance in up-to-the-minute design of hydraulically actuated machines.

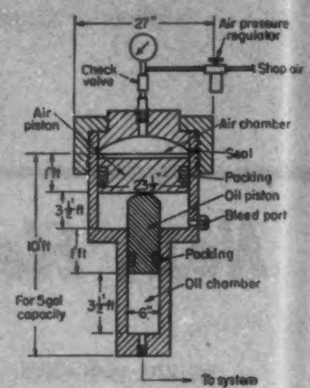
Separator type accumulators require no compressors or attachments to the gas chamber, nor is there any need for volume-discharge safety and regulation equipment. The mere elimination of this equipment compensates in many cases for the increased cost of multiple separator type accumulators.



Right — Simple form of piston type accumulator

Below—Floating piston type accumulator





Above — Differential piston type accumulator

INFLEXIBLE SEPARATOR TYPES

Separator type hydropneumatic accumulators can be divided into two categories: (1) Inflexible piston separator, and (2) flexible separator designs. In simplest form, the piston type resembles an actuating cylinder. High-pressure gas is precharged into the open piston end of the accumulator and hydraulic oil is then charged into the rod end. In this type accumulator, the rod is used only to indicate the volume of oil in the accumulator. Since indicators are seldom required, the floating-piston type is much more popular.

Floating Piston Designs: A typical floating-piston accumulator, developed for low-temperature operation in aircraft, consists of a honed cylinder into which is fitted a free piston containing two O-ring packings to seal the air from the oil. In order to maintain a differential pressure between the O-rings, a second piston is sealed within the floating piston. The smaller piston is back-loaded by Belleville spring washers. Low-temperature grease completely fills the chamber occupied by the Belleville washers and leads to an annular groove between the two O-rings. In this way, the pressure applied to the smaller piston works against the grease and is resisted by the Belleville washer, resulting in a pressure differential across each O-ring to provide adequate sealing.

Disadvantages of the piston-type accumulator are that it is costly to build and has practical limitations in size. Piston friction has a detrimental effect in pressure-regulator systems. It has been found that, over a period of time, oil leakage occurs, requiring that the unit be drained and recharged at intervals. Packings wear and require frequent replacement since normal leakage cannot be tolerated. The piston-type accumulator, when used as a surge chamber or shock absorber, has proved to be unsatisfactory. Inertia and acceleration factors of the piston itself intensify pressure peaks.

Differential Piston Designs: A differential piston type accumulator consists essentially of a large-diameter air cylinder mounted on top of a small-diameter oil cylinder, with the smaller piston butting upward against the air piston. The air chamber contains a fitting to which a shop air line or compressor is attached. With commercial differential piston type accumulators the air chamber is connected to an air tank and both units to an air compressor. The air tank makes it possible to keep the accumulator comparatively small as it allows complete exhaustion of the air. A unit having an oil capacity of 4½ gallons and a minimum pressure of 2000 psi weighs 4500 pounds, has an air tank weight of 8400 pounds and requires a floor space of 11 by 3 feet. A 10-gallon hydropneumatic accumulator with the same capacity weighs 170 pounds, is 9 inches in diameter and 4½ feet in overall length.

3—Separator Designs

By Edward M. Greer

President and Chief Engineer
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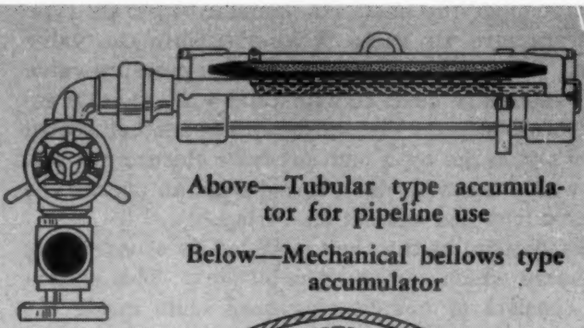
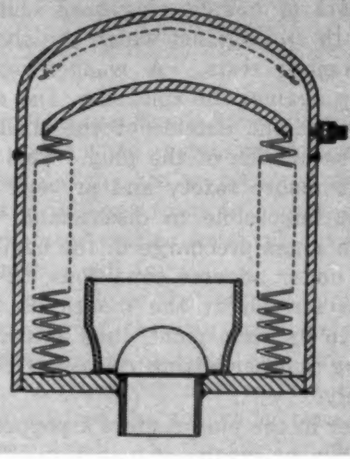


Above—Tubular type accumulator for pipeline use

Below—Mechanical bellows type accumulator



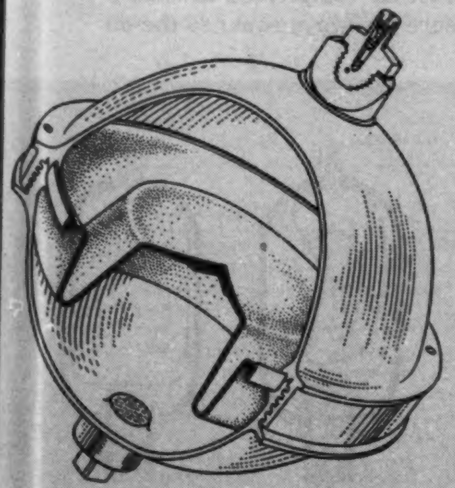
FLEXIBLE SEPARATOR TYPES

Most recent developments in the accumulator field have proved that flexible-separator type accumulators are the most practical from size, operational, and cost standpoints. These accumulators can be divided into the following types: (1) Bellows, (2) tubular-rubber, (3) rubber convoluted-diaphragm, (4) spherical, and (5) conical-bag.

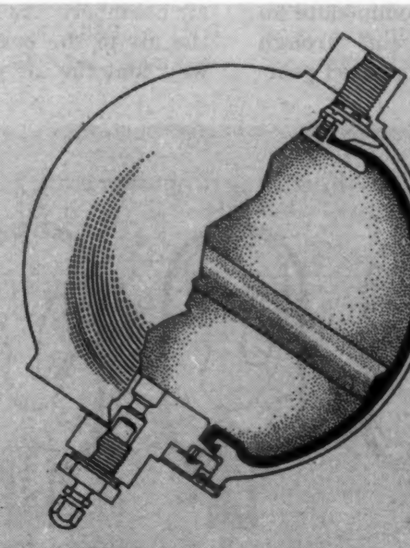
Mechanical Bellows Designs: This type unit consists of a dome-shape steel shell welded to a baseplate. A bellows is welded to the baseplate and is fully enclosed by welding the upper end of the bellows to a coverplate. An air valve is mounted on the wall in the upper end of the shell, and an oil inlet nipple is provided on the bottom of the baseplate. A tubular mechanical stop inside the bellows prevents collapse beyond a safe limit during air precharge.

One disadvantage of this design is that under high cycling conditions the bellows fatigue. It has also been extremely difficult to obtain a practical bellows that can be placed in a large chamber and precharged to high pressures. These accumulators, however, have proved fairly successful in limited applications as surge chambers or shock stops where the bellows has a short stroke.

Rubber Tube Designs: Tubular rubber accumulators have been designed primarily as surge dampers or desurgers. Such devices have been applied to oil pipeline systems by mounting in the discharge lines of single and multiple-stage reciprocating pumps. Generally, such units consist of a heavy-wall cylinder provided with end caps. A synthetic

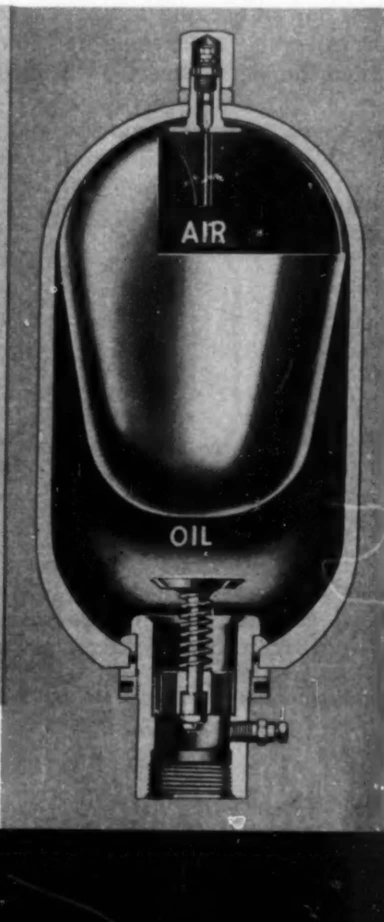


Diaphragm type aircraft accumulator



Above—Diaphragm action bag type aircraft accumulator

Right—Standard high-pressure bag type hydropneumatic accumulator



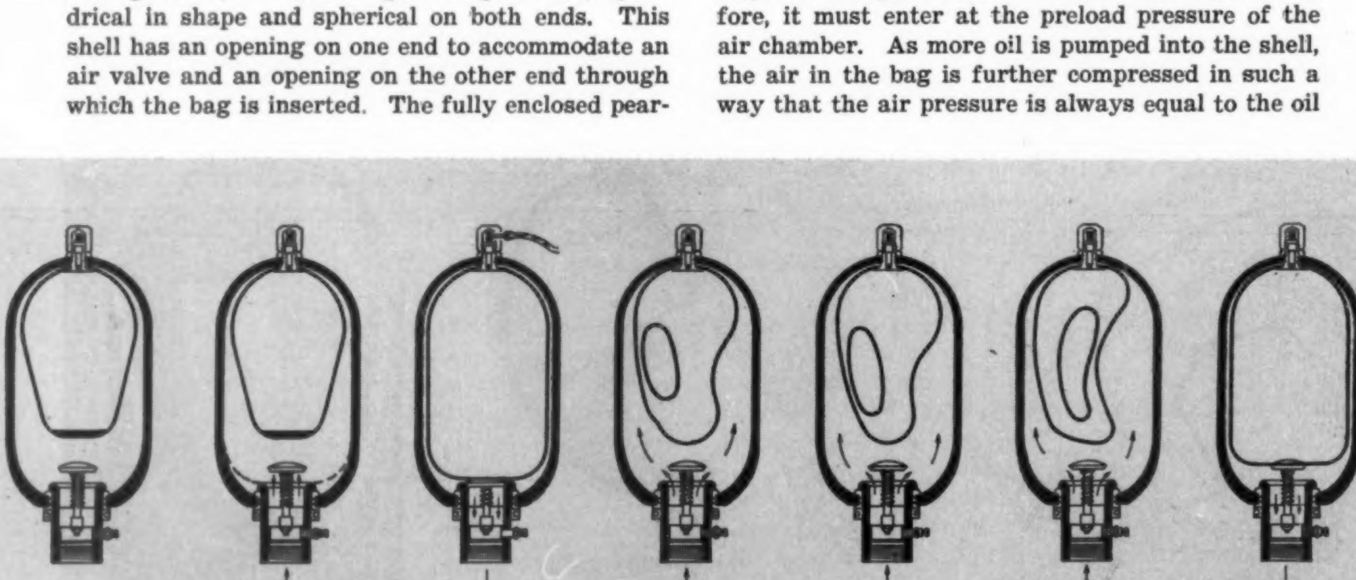
rubber tube is mounted axially in this cylinder, and is backed on its inside by a perforated steel tube which contains end fittings for connection to the pipeline. The rubber tube is sealed on the end caps to prevent leakage from the outside of the tube to the inside. Gas is precharged through a valve on the outer cylinder into the chamber around the outside of the rubber sealing tube. The perforated tube prevents the rubber tube from collapsing. As the pulsating oil is pumped through the inside of the unit, the pulsations are absorbed in the air chamber.

Diaphragm Designs: By far the greatest contribution to the development of diaphragm accumulators was made in the aircraft industry where high efficiency and low weight are essential. One typical aircraft accumulator is made of two hemispherical shells machined from chrome-molybdenum steel forgings. Acme threads lock the hemispheres securely clamping a convoluted diaphragm around the periphery. Another type of aircraft spherical accumulator consists of chrome-molybdenum drawn hemispherical shells welded at the periphery. In addition, the oil port and the end cap are welded to complete the shell. An open-end bag, placed in the shell, is clamped by a closure plug containing an air valve. A heavy welt molded around the horizontal diameter of the bag provides resistance to vertical bending, and prevents collapse around the periphery.

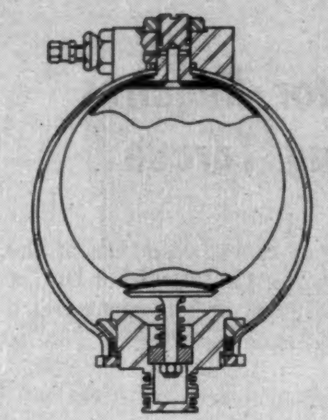
Bag Design Accumulators: Invention of the conical fully-enclosed bag type started a new era for hydraulic operation, especially in mobile and airborne equipment. This accumulator consists of a homogeneous, seamless, high-strength shell, cylindrical in shape and spherical on both ends. This shell has an opening on one end to accommodate an air valve and an opening on the other end through which the bag is inserted. The fully enclosed pear-shaped bag, made of synthetic rubber, is molded to an air stem. It is installed in the accumulator and mounted by means of a lock or jamnut on the upper end of the shell. A standard aircraft type high-pressure air valve is inserted into the valve stem. This air valve contains a high-pressure valve core, especially made to withstand a minimum pressure of 8000 psi. The valve core is further sealed after precharge by a high-pressure closure nut and the whole assembly is enclosed with an acorn-type cover nut sealed against an O-ring.

Bottom end of the shell is sealed by a novel plug assembly which contains the oil port. This assembly consists of a split ring that seats inside the shell mouth and against which the shoulder of the plug assembly rests. A washer and O-ring is backed up against the split ring, and a closure nut butts against the outside of the shell and screws against the outside of the plug. This construction is used to insure safety and prevent accidents by making it impossible to disassemble the accumulator with a gas precharge in the bag. To prevent bursting under adverse conditions, the shell is designed to stretch at the mouth as the metal is stressed to its yield point, thus extruding the rubber O-ring and permitting release of the fluid pressure safely.

A spider in the plug centers a poppet valve which is held open by means of a spring. The bottom of this poppet contains a piston and dashpot to effectively damp fast opening of the poppet under extremely high flow conditions. The accumulator is first preloaded with air or nitrogen at a predetermined pressure. Oil under pump pressure then enters the accumulator through the oil port. The first drop of oil which enters the accumulator must displace an equal volume of air in the bag; therefore, it must enter at the preload pressure of the air chamber. As more oil is pumped into the shell, the air in the bag is further compressed in such a way that the air pressure is always equal to the oil

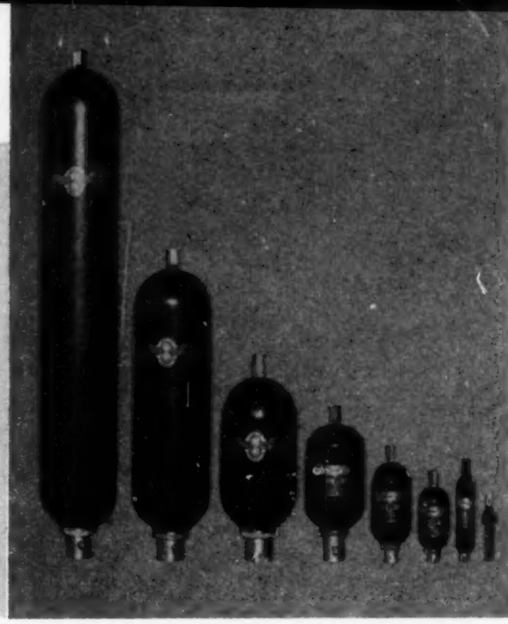


Normal cycles of the bag type accumulator: (a) Normal position; (b) injecting cushion oil; (c) precharging bag with nitrogen or air; (d) charging from pump, oil-air ratio of 3-to-1; (e) charging from pump, oil-air ratio of 4-to-1; (f) charging from pump, oil-air ratio of 5-to-1; and (g) discharging oil to system



Left — Spherical version of bag type accumulator

Right—Size range of bag accumulators



pressure, and the bag floats in equilibrium. Inasmuch as oil under usual system conditions virtually is incompressible, the air or nitrogen under pressure acts as a dynamic force to maintain pressure in the accumulator and to force oil out when system requirements call for it.

Operating principle of the bag is simple. It will be readily seen that the bag will stretch around the top at its greatest diameter and thinnest wall, and will gradually stretch downward and push the bag outward against the sidewalls, thus squeezing out the oil completely. In the ideal or molded condition the volume of air is one-half the total volume. It has, however, been found very practical to compress the air in the bag at a ratio of 5-to-1, and what happens in a normal cycle is shown in the accompanying series of drawings.

To meet various industrial requirements, this type of accumulator is produced in the sizes ranging from 2 cubic inches to 10 gallons. Larger capacities are secured by connecting multiple units in any desired combination. Two pressure ranges are available—a medium-pressure series, 1000 to 1500 psi, and a high-pressure series of 3000 psi. For special applications, units ranging from 50 psi to 10,000 psi operating pressure volumes from 1 cubic inch to 100 gallon capacity have been put into satisfactory service. These capacities are full air volume, with the bag fully extended and no oil in the accumulator. A spherical accumulator recently developed along the same lines consists of a seamless spherical chrome-molybdenum steel sheet without welds or joints. Following the principle of conical bag accumulator, the bag—molded fully enclosed on its air stem—is inserted into the shell and held in position by a suitable locknut. A closure plug contains a spring-loaded poppet which impinges against the bag and keeps it off the port opening until the oil is fully exhausted. The bag wall thickness increases gradually and follows the same Olaer patents and principles as in the conical-bag accumulator.

To permit overall evaluation of accumulator types by characteristics, the accompanying chart provides comparative ratings. Selection can be made according to design requirements of the particular system contemplated. Advantages of the bag types are self-evident from the code ratings.

Evaluation of Accumulator Types According to Characteristics*

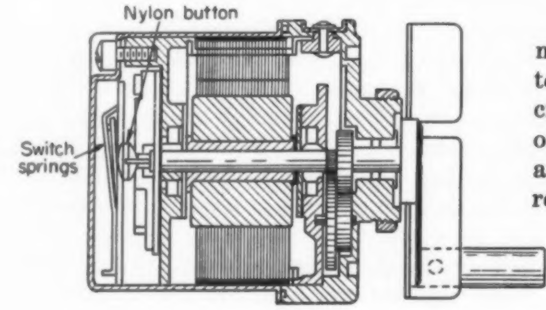
					Flexible Separator Types			
Type of Accumulator	Weight Loaded	Spring Loaded	Non-Separator	Floating Piston	Bellows	Diaphragm	Tubular Rubber	Enclosed Bag
Cost vs. capacity	7	5	1	5 to 7	5	1	4	1
Weight vs. capacity	6	5	1	4	5	1	5	1
Scope of application	7	5	1	1	2	1	1	1
Size limitations	1	7	3	5	6	5	6	3
Amount of extra equipment required	2	1	7	1	1	1	1	1
Ability to vary pressure	3	5	3	3	4	3	3	3
Pressure control through cycle	1	6	5	5	6	5	5	5
Space required	6	6	1	1	2	1	1	1
Maintenance frequency	6	6	1	5	6	6	4	3
Precharge loss	1	3	7	6	1	3	3	1
Fluid leakage	3	2	1	6	1	1	2	1
Replacement parts availability and cost	6	6	1	2	7	3	5	4
Life expectancy	1	5	1	4	7	6	3	3
Safety features	1	1	6	5	2	6	6	2
Operational safety	1	1	2	2	2	2	2	2
Efficiency	1	3	6	1	4	3	3	1
Installation ease	6	1	4	2	2	2	2	3

*Code: 1—Excellent, 2—Very good, 3—Good, 4—Fairly good, 5—Fair, 6—Poor, 7—Very poor

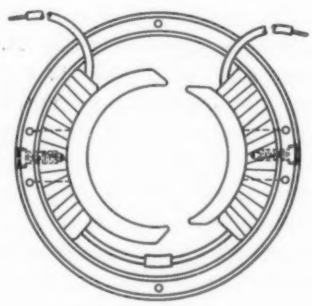
CONTEMPORARY DESIGN

Hand Generator Requires Low Starting Torque

SAID to be unique in hand generators, a feature of the new RWG-2308 hand generator of Holtzer-Cabot Div. of National Pneumatic Co. is a pole shoe arrangement which limits starting cranking torque to a maximum of 2½ lb-in. For use in field telephones and switchboards, radio control units and other communication devices, and test sets, the generator meets U. S. Signal Corps requirements and is mechanically and electrically interchangeable with other types of generators. Approximately 2 9/16 inches in diameter and 2 15/32 inches long, the generator produces an open-circuit voltage of 95 to 100 at 1.75 watts and 20 cycles ac with a 200 rpm crank speed.

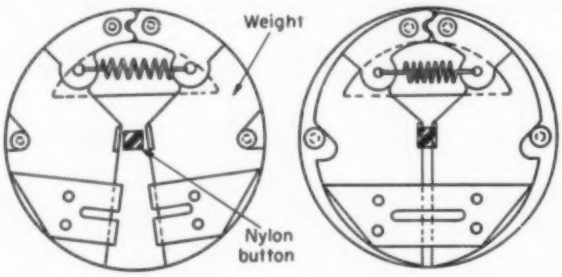


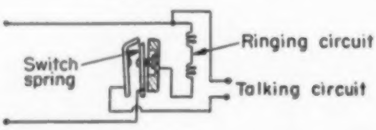
Generator is essentially a two-pole Alnico permanent magnet type, with the Alnico rotor fastened to the knurled shaft with babbitt. A speed increaser with 1 to 6 ratio provides a rotor speed of 1200 rpm with 200 rpm crank input. Bearings are phosphor bronze with oil feed from a built-in reservoir through felt wicks.



Asymmetrical pole shoe arrangement is responsible for the low starting cranking torque. Nearly all "cogging" and variable-torque characteristics are eliminated by the arrangement, which maintains constant reluctance in the magnetic circuit.

Acoustical shock to the telephone receiver is prevented by allowing the line to discharge its electrical energy (from ringing) before the talking circuit is established. A double-pole cut-in switch, actuated by a nylon button, makes the change. Acceleration of the rotor shaft causes two governor weights to open, permitting a switch spring to make the ringing contact. As the shaft decelerates, a spring pulls the weights together, and wedge-shaped faces on the weights cause the nylon button to return the switch spring to contact with the talking circuit. Instantaneous closure is prevented, however, by a dashpot which introduces a time delay of approximately ½-second.





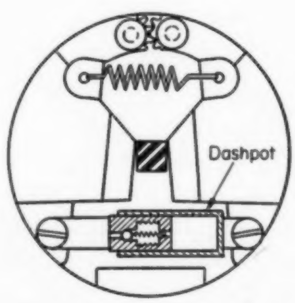
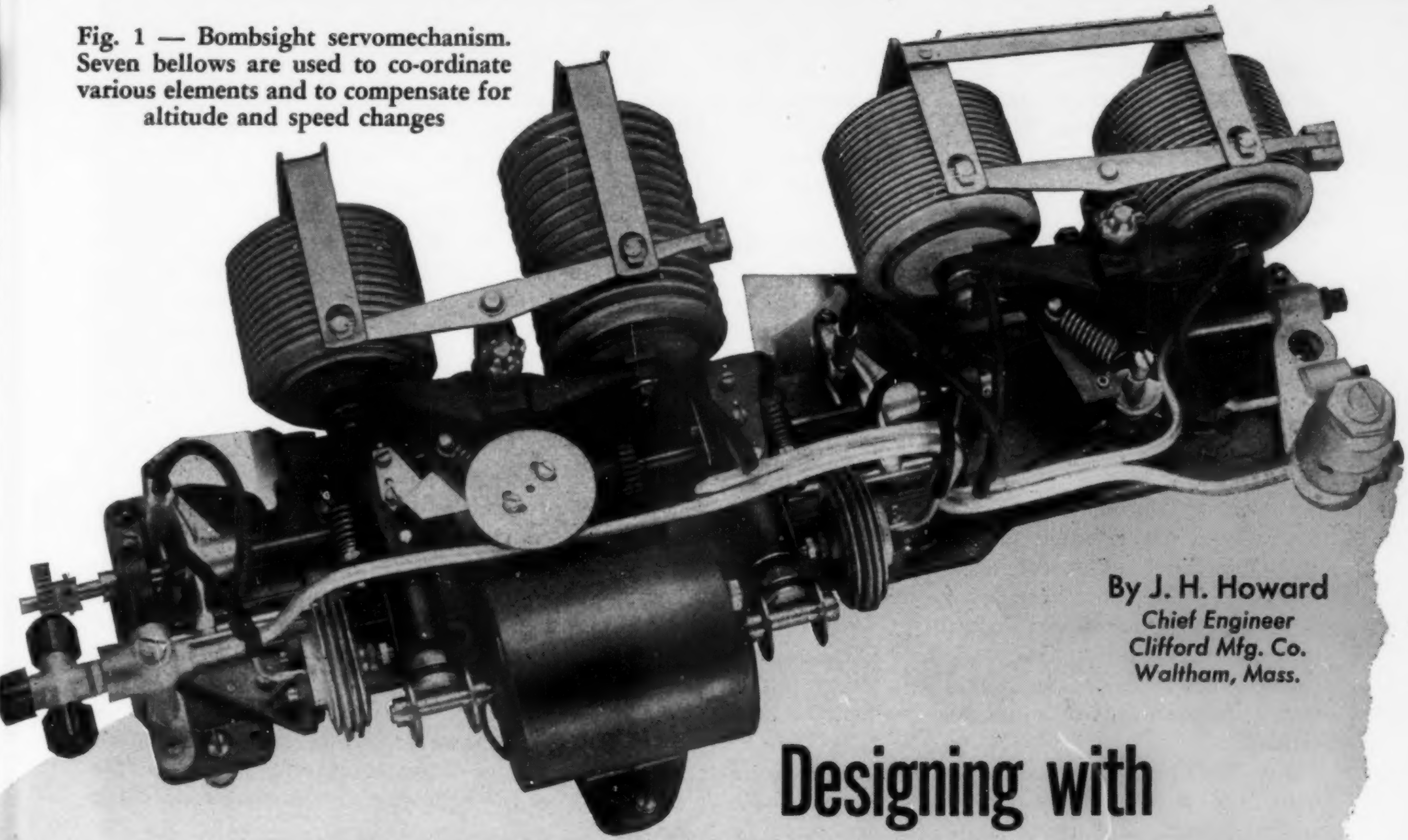


Fig. 1 — Bombsight servomechanism. Seven bellows are used to co-ordinate various elements and to compensate for altitude and speed changes



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Designing with METAL BELLOWS

Factors in their selection for temperature and pressure control or indication, motion transmission, shaft seals and other mechanical uses

POSSESSING a rather unique versatility in design, metal bellows have long been used as pressure and temperature-sensitive elements. But besides serving for control or regulation in thermal or pressure systems, *Figs.* 1 and 2, they have also proved effective for motion transmission, for seals around rotating shafts, and for flexible piping.

A metal bellows might be thought of as an axially elastic pressure vessel. It is analogous in behavior to a helical spring, operating in compression, and it is also relatively elastic in flexure. Extension beyond free length is not recommended. For control and/or indication, a principal virtue of bellows is that they respond with appreciable motion and force. Generally, for actuation of the

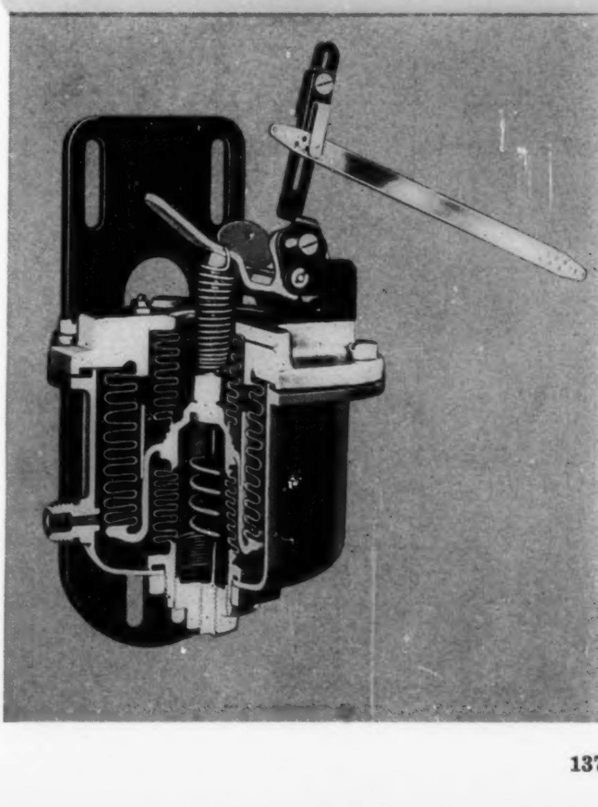


Fig. 2—Right—Bellows assembly for Taylor Instrument Co. ratio controller. Such instruments are used to control or indicate temperature, pressure, rate of flow, liquid level, etc., according to a desired ratio or differential with another variable

next element in the control series, such as a valve or switch, no complex auxiliary apparatus is required for amplification of either force or motion. Bellows are commonly available as standard components in a wide range of sizes with a variety of end designs and fittings.

This article summarizes principles of different applications of bellows, the proper selection of bellows materials, determination of bellows proportions, basic types of bellows assemblies, and assembly methods as they influence bellows specification.

Design for Temperature Control: Use of bellows as thermostats is well known by their common application in automotive engine cooling systems where they are situated in the flow path of the cooling medium and operate an integrally assembled valve. However, as thermostats, their most prolific use is for the remote control of temperature, proved in such applications as refrigerators, ovens, hot-water storage tanks, processing tanks, wax pots, solder baths, and various heating specialties.

For this type of service the bellows is connected with flexible capillary tubing to a thermal bulb which extends into the chamber or space to be regulated. The entire bellows, tube and bulb assembly is filled with the necessary thermo-sensitive gas or liquid. Variations in the temperature of the space to be regulated effect corresponding changes in the contained gas or liquid. These changes are transmitted through the tubing to the bellows which extends or retracts, actuating the switch or valve which governs the heat added to or subtracted from the space to maintain or restore the desired temperature.

For different circumstances and temperature ranges, the necessary volume change is obtained from (1) expansion of an inert gas, (2) variation in vapor pressure of a volatile liquid, or (3) thermal expansion of a nonvolatile liquid. Generally, these different physical principles are called into play according to the temperature being sensed or the magnitude of force required of the bellows.

GAS EXPANSION: The gas-expansion principle is used in specialized controls which draw upon its ability to function at much higher temperatures than can be withstood by the vapor-pressure and liquid expansion type. For example, in a pilot flame control, an inert gas, such as helium, may function in the 900-1000 F range.

For temperature under about 600 F when close control is required this type is not suitable. Even with a relatively large-volume bulb and a small-volume bellows, motion and force are only slight. With a temperature differential of as much as 15 or 20 F, the volume change is so small that almost complete freedom from friction losses in the controlling device itself is required for operation. In a pilot control, however, temperature differential is unimportant and thrust of only a few ounces is required of the bellows to actuate the controlling device.

VAPOR PRESSURE: The vapor-pressure principle is quite widely used at temperatures both above and below normal room temperatures. When a confined liquid is heated, vapor pressure is generated according to definite and predictable curves, *Fig.* 3, and is transmitted through the capillary tubing to the bellows, *Fig.* 4.

In the design of vapor-pressure systems, several pertinent factors must be considered: temperature range, required sensitivity and upper pressure

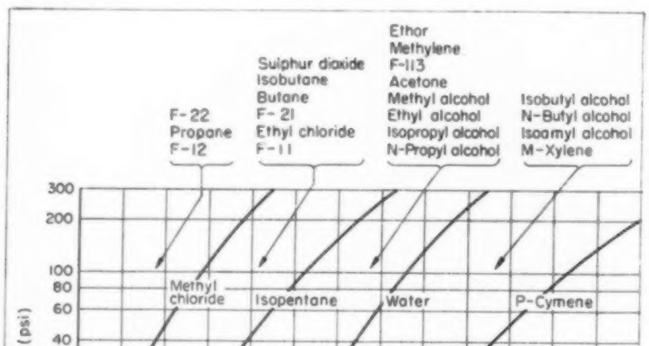


Fig. 3—Left—Vapor pressure versus temperature for a few liquids used in vapor-pressure type bellows thermostats. Additional liquids are listed above the curves according to their approximate order

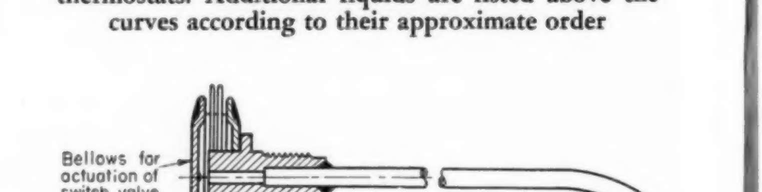


Fig. 4—Above — Typical bellows, capillary tubing and bulb assembly for vapor-pressure thermostatic operation

limit. These requirements must be met by the right combination of the volatile gas, sizes of bulb and bellows, and "fill" of the system.

Differentiation in practice occurs according to whether the control is for temperatures below normal room temperature, above normal, or for both. For example, a control to operate at zero F must be charged with a thermal liquid having an active vapor pressure at that point, although the boiling point may be either below or above the control point. Methyl chloride would be a logical filling medium; it has a boiling point just under minus 10 F and a vapor pressure of 3 to 4 psig at zero F. However, its vapor pressure at 120 F is about 150 psi. When used in refrigeration equipment, this filling medium performs the control function satisfactorily, but produces excessive pressure for the bellows when there is no cooling as during shipment or storage of the unit. Two or three-ply bellows can be employed, but a more economical solution is to modify the temperature versus pressure curve of the bulb and bellows system.

Such modification is introduced by limited filling of the system. That is, a volume of volatile liquid much less than the volume of the system, is so determined that if it will completely volatilize at a temperature just above the maximum desired control temperature. Beyond that point, then, only gaseous expansion occurs at a relatively minute rate, *Fig.* 5.

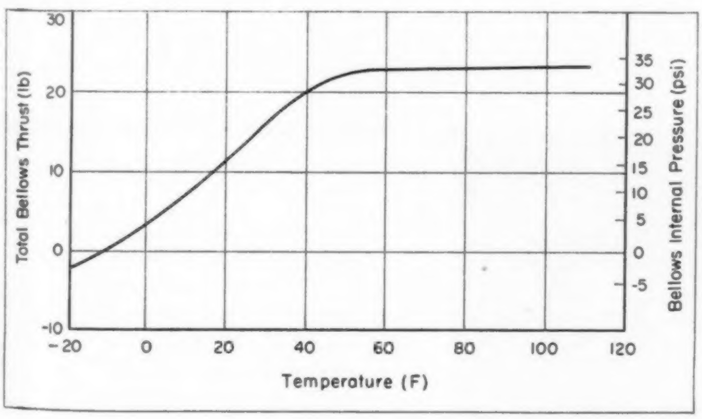
Ambient temperature has no significant effect on "limited-fill" controls because the volatile liquid is contained in the bulb, which generally may be quite small. Expansion of the superheated fluid in the tubing and bellows is negligible.

For service above room temperature, the system is entirely filled with a volatile liquid having a boiling point above or below but in keeping with the control temperature. If the boiling point is below the control point, as temperature increases, the liquid in the bulb volatilizes and the expansion is transmitted hydraulically, as it were, by the liquid in the tubing to the filled bellows. Actually, only a portion of the liquid in the sensing bulb is volatilized.

In these systems the volume of the bulb definitely determines the amount of bellows stroke. The bulbs are designed to trap the vapor so that only liquid is forced into the tube. Irregular operation results if liquid and vapor are allowed to mix. Overtravel protection is provided inherently by limitation of bulb size. Controls of this type usually are unaffected by ambient temperature since only liquid expansion or contraction at a minute rate is involved in the tubing and bellows.

If there is selected a liquid having a boiling point above the control point, the system then operates under vacuum up through the control point. Vacu-

Fig. 5—Below — Curve of temperature versus thrust for "limited-fill" $1\frac{1}{8}$-inch bellows operating with methyl chloride. Volatilization of the liquid is completed at about 50 F; only gaseous expansion occurs above 50 F



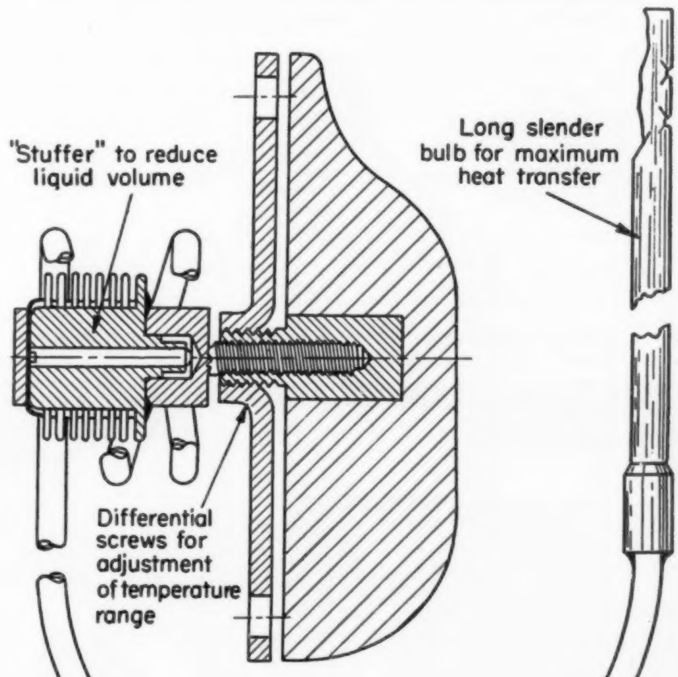


Fig. 6—Right—Typical liquid-expansion type bellows thermostat assembly with adjustment screws

um usually should be limited to about 15 inches of mercury. The virtue of vacuum operation is a "fail safe" feature in some applications, such as automotive thermostats, in the event leakage occurs in the bellows or elsewhere in a closed system.

Universal Service: At temperatures both below and above room temperature universal systems can be designed that draw upon principles of operation of both the low and the high-temperature types just described. The bellows, capillary and bulb assembly is charged with a definite volume of the thermal liquid, and the bulb is so proportioned that it can contain all the liquid. When the capillary and bellows, with the latter in an extended condition, are completely filled with the thermal liquid there must still be enough left in the bulb to generate necessary vapor pressure.

With the bellows and capillary higher than the operating temperature at the bulb, liquid in the bellows and the capillary is volatilized and subsequently condenses in the bulb. The bellows is filled with a superheated gas and receives its impulses from the bulb through this gas. A so-called transition point is reached when the temperature of bellows and capillary reaches that of the bulb. As the temperature of the bellows and capillary drops below that of the bulb, the liquid in the bulb rapidly volatilizes and condenses in the bellows and capillary until they are completely filled. Thermal impulses are then transmitted hydraulically.

This type of control is, of course, more expensive because special measures must be taken in the accurate filling. Bulbs are also of necessity considerably larger. Moreover, a heavy or two or three-ply bellows may be needed to withstand the vapor pressures incident to higher temperatures encountered in shipment or storage. Despite these relative disadvantages, the universal type control satisfies requirements that can be filled by neither the low-temperature nor high-temperature vapor-pressure types alone.

Universal operation is also possible with a "dual-fill" system. One of the charging media is a high boiling point liquid which acts as the hydraulic fluid between the bulb and the bellows. The other is a small amount of volatile liquid, insoluble in the transmission liquid, placed in the bulb. The capillary tube terminates in the center of the bulb, and the globule of volatile liquid and its gas are trapped in the bulb, regardless of bulb orientation. This type can be a limited-fill system and/or a fail-safe system.

Design Considerations: In all of the vapor-pressure controls, bellows sizes generally range from

Table 1—Coefficients of Thermal Expansion

	Coefficient* (in.³/in.³/deg F) × 10³ (0 – 1 – 2)	Freezing Point (F)	Boiling Point (F)	Max. Op. Temp. (F)
Mercury		−38	675	
H. T. A.†			672	600
Glycerine			554	350
H. T. C.†		29	528	700
H. T. B.†		54	500	700
Trichlorobenzene		45	415	450
O-Toluidine		3	392	450
Ethylene Glycol		11	388	350
Aniline		−22	364	450
O-Dichlorobenzene		2	357	450
Chlorotoluene		−35	321	450
Turpentine			311	350
Xylene		−53	283	400
Monochlorobenzene		−50	269	450
Glycerine-Water (62:38 vol.)		−48	235	350
Toluene		−139	231	400
Water		32	212	
Trichloroethylene		−99	188	250
Benzene		42	176	500
Ethyl Alcohol		−174	173	250
Carbon Tetrachloride		−9	170	250
Methyl Alcohol		−143	148	300
Chloroform		−82	142	250
Acetone		−138	133	250
F-113		−31	118	250
Methylene Chloride		−142	104	300
Ethyl Ether		−177	94	250
F-11		−168	75	250
Ethyl Chloride		−218	55	350
F-21		−197	48	250
Methyl Bromide		−137	38	250
Butane		−211	31	400
Sulphur Dioxide		−105	14	800
Isobutane		−229	10	400
Methyl Chloride		−148	−11	400
F-12		−247	−22	250
Ammonia		−107	−28	
Propane	2.4→	−310	−44	400

*Values increase with temperature; for example, coefficient for trichlorobenzene ranges from 0.00044 in the region of its 45 F freezing point to 0.00070 at its recommended maximum operating temperature, 450 F.

†Special high-temperature service compounds.

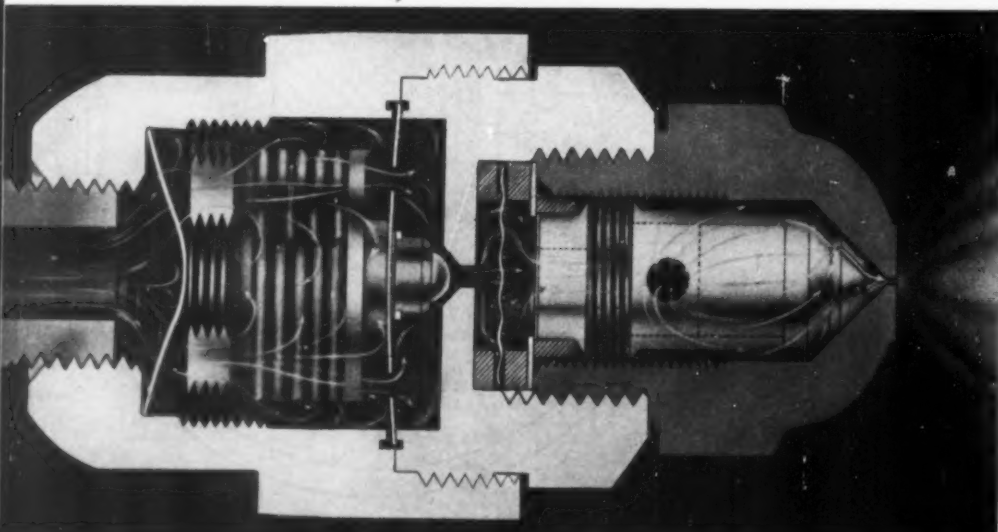


Fig. 7—Minneapolis-Honeywell bellows-controlled ball-valve design for oil burners. Located between the sealed bellows and the ball, a snap disk is actuated by the bellows, according to fuel oil pressure, and permits the ball to retract or forces it against its seat

11/16 inch to 4½ inches, according to required sensitivity and stroke. Smaller size bellows are not satisfactory for this type control, since they lack required flexibility.

Vapor-pressure thermostats have the advantage of high sensitivity and, if necessary, relatively long stroke. They have the disadvantage of being affected by variations in the force or friction which they must overcome in the normal operation of valves or switches for which they are designed and are also affected by barometric pressure changes.

Thrust of the bellows at any temperature is definitely fixed by the vapor pressure of the filling medium and cannot be changed. Similarly thrust differential between any two temperatures is fixed by the filling medium.

Therefore, in the design of switch or valve mechanisms, friction should not only be minimized but also be maintained constant. Also, to compensate for fluctuations in pressure against which a bellows-operated valve must function, a balanced valve such as the double-seated or piston type must be used.

Another factor to be considered is the adjustment means. Usually, for adjustment of the temperature range, a spring of proper proportion and a take-up means, such as a screw, are provided. Rate of the adjusting spring must also be held as low as possible because the operating differential of the device will be governed largely by this factor. If the spring aborbs an unduly large portion of the force available from the filling medium, the residual force may be inadequate and a wide operating differential results.

In design, these various factors can be combined and analyzed by the following approximate equations which yield reasonably accurate results:

$$T_d = \frac{F_t + sk_b}{P_v A}$$

or

$$T_d = \frac{F + s(k_b + k_s)}{P_v A}$$

where T_d = operating differential of switch or valve, degrees F; F = differential force required actually to operate the switch or valve *excluding* the adjusting spring, pounds; F_t = differential force required actually to operate the switch or valve *including* the adjusting spring, pounds; s = stroke to operate valve or switch, inches; k_b = spring rate of bellows, pounds per inch; k_s = spring rate of adjusting spring, pounds per inch; P_v = change in vapor pressure of filling medium at approximate operating temperatures of thermal bulb, psi per degree F; and A = effective area of bellows, square inches.

Differential forces F and F_t are, in each case, the difference between the force at the start and at the end of the operating stroke of the switch or valve. Force F would be determined by removing the adjusting spring and measuring the actual force at both ends of the operating stroke under actual operating conditions. Force F_t is determined the same way except that the adjusting spring is in place and the valve or switch is complete, excluding the bellows itself, of course.

The equations give fairly accurate results with a switch or snap-action valve. However, with a modulating type valve, the actual operating differential would be considerably less. Stroke s is, of course, the full lift of the valve, but in operation the valve "floats" and does not travel the full distance from closed to wide open.

THERMAL OR LIQUID EXPANSION: Utilizing the thermal expansion of an incompressible liquid, liquid-expansion designs are completely filled, and changes in volume resulting with changes in temperature at the bulb are conveyed hydraulically to the bellows, *Fig.* 6. Because thermal expansion rates are quite low, these systems necessarily utilize the smaller sizes of bellows, such as 15/32 or 9/16-inch, in order to obtain significant bellows stroke. Conversely, the bulb must be relatively large, particularly so for maximum sensitivity. As TABLE 1 shows, many filling liquids are available, giving latitude for the choice of expansion coefficient to suit a desired temperature range.

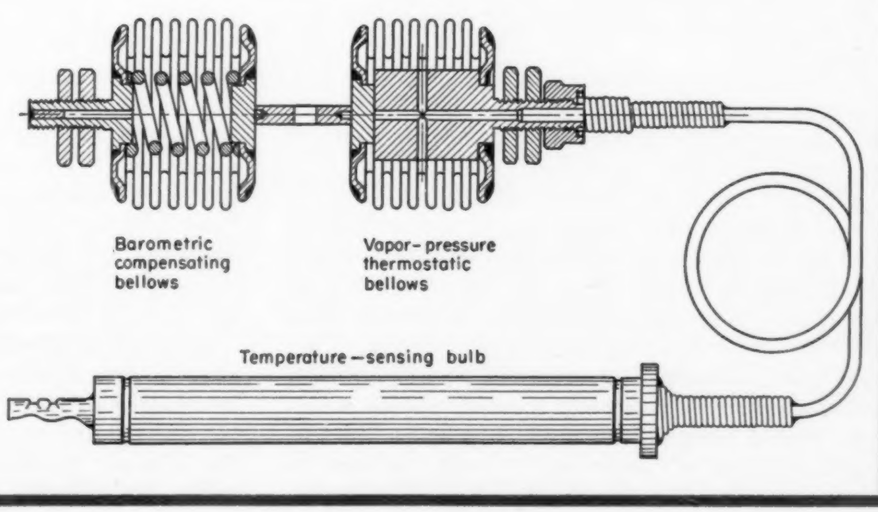


Fig. 8—Bulb and bellows thermostatic assembly with barometric compensation

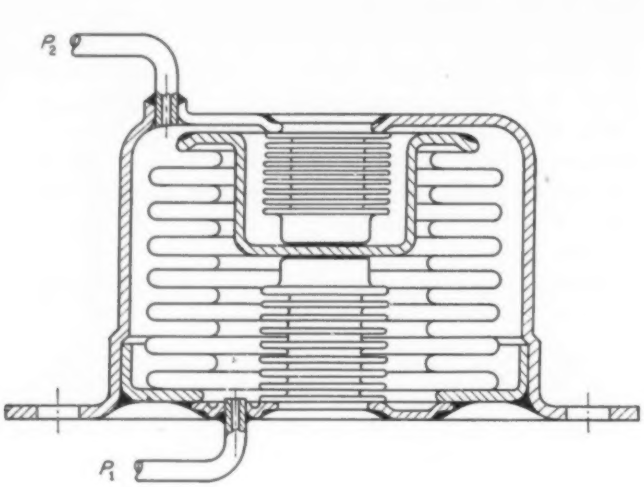


Fig. 9 — Differential-pressure bellows assembly. Movement of actuating rod is measure of difference between pressures P_1 and P_2

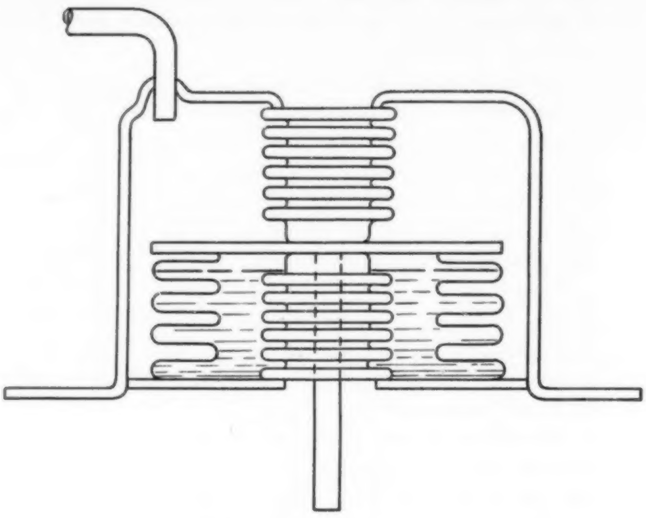


Fig. 10—Bellows assembly for overpressure protection. Liquid in large bellows serves as a stop at selected maximum pressure

Obviously, this type is limited in application to valves or switches requiring only small movement. The majority of successful applications use switches requiring only 0.001 or 0.002-inch movement of the bellows and in no case more than approximately 1/32-inch. Valves usually are of the modulating typ- with movement to 1/16-inch. Lever arrangements can be used, however, to obtain greater movement.

Maximum movement of the bellows is usually about 0.003-inch per degree F for a 9/16-inch diameter bellows with 26 convolutions, and 0.006-inch per degree F for a ¾-inch bellows with 28 convolutions. These maximum sensitivity sizes are restricted to a temperature adjustment range of about 60 F, say between 100 and 160 F or 220 and 280 F.

If a wider range of adjustment must be accommodated, sensitivity will be reduced correspondingly. In domestic stoves, for example, the temperature adjustment range is usually 300 or 400 F. Through special design of switches and valves, movement is only approximately 0.0002-inch per degree F.

Bulb design is critical from the standpoint of obtaining maximum heat transfer. It should have as small a diameter as possible and correspondingly the length becomes a maximum.

Required use of small sizes leads to the need for return springs with liquid expansion bellows. Otherwise the bellows will not follow the action of the filling medium, particularly on the cooling leg of the cycle. Atmospheric pressure does not overcome the natural spring force of the bellows. This condition is especially apparent for designs in which liquid pressure is internally applied. Since a bellows should normally be used only below its free length, it must be compressed to locate the operating range below the free length.

With the bulb at the lowest operating temperature and the bellows at its shortest length, the opposing spring should exert no less than 10 pounds thrust on the ¾ or 9/16-inch bellows. The maximum pressure during the operating cycle should be less than 20 pounds for the ¾-inch size, less than 25 or 30 pounds for the 9/16-inch size. Overtravel protection is usually provided by having the spring of such proportions that its force is still not excessive beyond the operating stroke and it does not compress solidly for some distance beyond the normal stroke.

However, if the bellows is enclosed in a cup and pressure is applied externally, the pressure tends to compress the bellows in the desired direction, and the spring rate can be reduced to the extent of the spring rate of the bellows itself. The cup arrangement is frequently advantageous because overall forces in the system are less and, hence, friction is reduced.

No range adjustment can be made by varying the spring load as in the case of the vapor pressure type. Instead, it can be adjusted by changing the clearance between the movable end of the bellows and the switch or valve, or by changing the anchor point or fixed end of the bellows.

Thermal expansion bellows are adaptable over a wide range of temperature and are compact. Regulation is possible from about −100 F to 650 F. They also, by their nature, generate high thrust force and are not affected by reasonable variations in the frictional and functional characteristics of the valve or switch being operated.

Ambient temperature can have an appreciable effect if care is not exercised in design. Error can be reduced usually to negligible limits by minimizing the volume of liquid in the capillary tube and the bellows. Smallest possible bore tubing must be used. An internal guide or "stuffer" can be de-

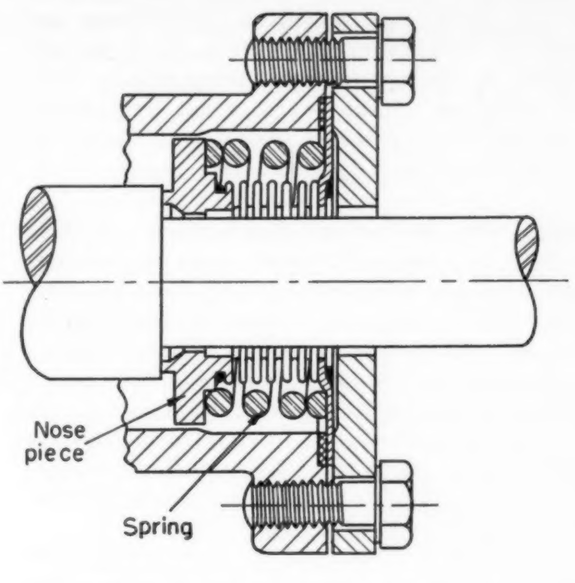


Fig. 11—Typical bellows shaft seal

signed for the bellows to displace a major portion of the liquid otherwise filling the bellows.

The expected operating temperature differential of the controlled switch or valve can be calculated from

$$T_d = K \frac{sA}{CV}$$

where T_d = operating differential of switch or valve, degrees F; K = a constant depending upon the flexibility of the bellows (K = 1.25 is satisfactory for ordinary small bellows); s = bellows stroke to operate valve or switch, inches; A = effective area of bellows, square inches; C = coefficient of cubical expansion of the filling liquid, cubic inches per cubic inch per degree F; and V = volume of bulb, cubic inches.

The formula is quite effective for a snap-acting switch or valve. For a modulating valve, if the stroke is taken as the total travel from closed to wide open, the actual operating temperature will be very much less.

Design for Pressure Sensing and Transmission: Previously described thermostat applications of bellows, since they are in reality pressure uses, have already suggested some of the possibilities for strictly pressure service.

Bellows function with pressure applied internally or externally. They can be used in liquid or gas systems equally well. Direct control or indication of pressure is possible from a few psi to more than 3000 psi. As noted in thermostatic applications, however, wide-range operation is accompanied by low sensitivity. In many control applications, though, the pressure range is usually small enough to permit adequate sensitivity or only a particular maximum or minimum pressure is involved. Details pertaining to design for pressure and stroke ranges will be discussed later on in this article.

Direct-pressure design analagous to direct thermal control in automotive thermostats can be employed. The oil-burner ball-valve control arrangement shown in *Fig.* 7 demonstrates this use. Fuel oil pressure is applied externally to a sealed bellows. At the desired operating pressure, the bellows is compressed sufficiently to actuate the snap disk and open the valve. When no fuel is applied, the bellows returns the ball to its seat.

Another basic use occurs in devices requiring barometer compensation. For example, aircraft fuel-air ratio is frequently controlled by a bellows. The bellows is evacuated and sealed, and expands, of course, as atmospheric pressure decreases with higher altitudes. Thus, the fuel and air mixture is automatically regulated by altitude.

Still another use on aircraft occurs in conjunction with vapor pressure thermostats, *Fig.* 8, earlier described. When a bellows is used for temperature control with the vapor pressure principle, its success on aircraft depends upon a means for offsetting the reduced external pressure at altitude. Such correction is quite simply and accurately provided by an evacuated bellows mounted in line with the vapor pressure bellows. Reduced external pressure on the evacuated bellows permits it to expand and counteract the corresponding tendency of the "temperature" bellows.

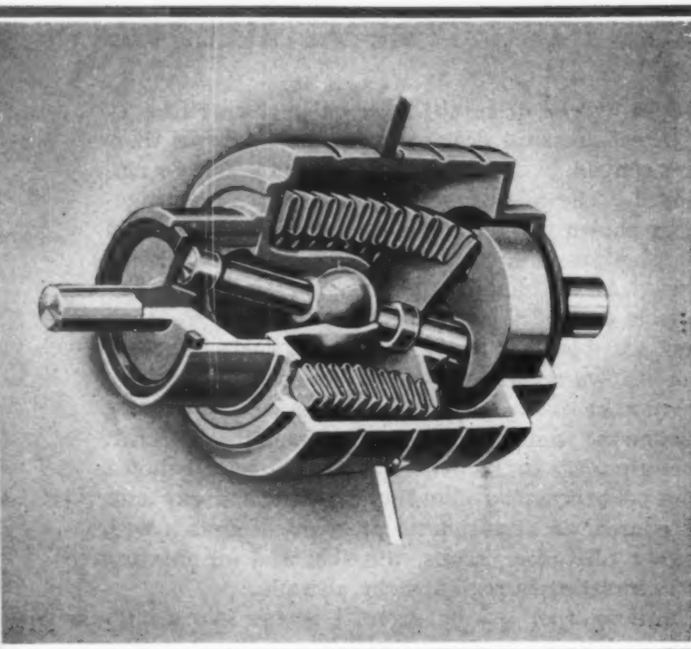
Bellows also are applied in process control or measurement in several ways. One is their use with a Venturi for "sensing" flow. The difference in pressure detected at two points in the Venturi by bellows is a function of the flow rate.

Of relatively recent origin, the differential-pressure bellows assembly is one of the most effective devices for direct sensing of pressure differences with high sensitivity, *Fig.* 9. The assembly consists of a large expanding bellows sealed at both ends by two smaller bellows situated within the large bellows. The large bellows is then so contained within a cup or housing that pressure or vacuum can be applied to it internally and externally. Movement of the actuating rod is proportional to the difference between the pressures on the inside and outside of the bellows ($P_1 - P_2$). Either side of the large bellows can be evacuated and sealed. Then the movement of the rod is a function of absolute pressure.

Greater latitude in design and usually high sensitivity can be gained with the differential-pressure bellows as contrasted with the use of two separate bellows.

A variation of the differential-pressure bellows is shown in *Fig.* 10. The main bellows of this unit is partially filled with a liquid to serve as a stop if pressure exceeds a desired limit. The unit is especially intended for absolute pressure service. It is very sensitive at low pressures but will withstand high pressures.

A simple but quite versatile service of bellows is for motion transmission or multiplication — hy-



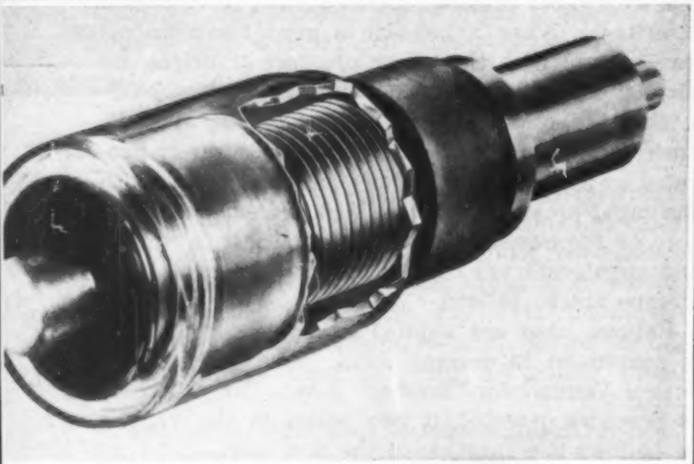


Fig. 12—Three types of bellows seal applications. Top, hermetic flexible bellows seal designed by the Kearfott Corp. for transmission of rotary motion from exterior to interior of high-vacuum systems. Center, Jennings Radio Mfg. Co. vacuum capacitor with a variable plate controlled through a bellows seal. Above, Watts Regulator Co. low water cut-off in which bellows transmits angular motion from float to mercury switch

draulically. If two or more bellows are connected by tubing, and the whole assembly liquid filled, force and movement applied to one bellows are transmitted throughout the system in a manner analagous to the operation of hydraulically connected cylinders. Ratio of the input and output movements can be selected over a wide range by proper specification of bellows sizes.

Design for Mechanical Applications: Besides their uses in temperature and pressure control, bellows serve as seals, flexible piping, and couplings.

Bellows shaft seals are used as packing glands to prevent leakage around rotating shafts ranging from 5/8-inch upward in diameter. Their major advantage is on special applications where an all-metal seal is necessary. These applications are generally those where high temperature is encountered or where liquids or gases used would adversely affect organic seal materials.

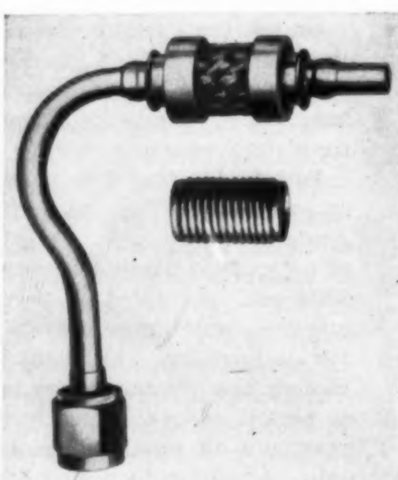
For rotating shaft seals, *Fig.* 11, the bellows is joined to a nose piece which is the actual sealing member in contact with an opposed surface. Any seal face material can be used, such as bearing bronzes or carbon, ceramics, etc., although fully leakproof assembly of some nonmetallic nose pieces to bellows may be difficult to obtain.

Force on the seal face can be maintained constant and independent of pressure by making the effective area of the bellows equal to the equivalent area of the nose piece. In other words, the mean diameter of the bellows should equal the mean diameter of the nose piece. Usually a spring is used in conjunction with the bellows to maintain the desired force at the contact plane.

Other seal applications are also served by bellows. Axial or flexural movement of the bellows is usually involved. Several of these uses are shown in *Fig.* 12. Packless valves and steam traps are still other places of application. As axially and laterally flexible pressure vessels, bellows are applied in all types and sizes of piping system as expansion or flexible members. They are particularly used to allow for thermal expansion and to simplify alignment problems. A primary example is the jet engine flexible manifold shown in *Fig.* 13.

As flexible joints, bellows are applied in per-

Fig. 13—Flexible jet-engine manifold assembly. Stainless steel bellows withstands high temperature, accommodates thermal exsion, prevents leakage, and simplifies alignment



haps their greatest size range. They are normally available in diameters from 7/8-inch to 30 inches, and much larger sizes have been produced for special applications.

The foregoing discussions have summarized the principal applications of bellows. However, there are other demonstrated specific uses which may offer possibilities in certain design areas. For light services, such as in servos, bellows are used as mechanical shaft couplings, minimizing need for precise alignment. Applied in a manner like springs, they are also used for shock mountings or vibration dampers.

Bellows Selection: Three principal considerations influence bellows selection: bellows materials, bellows proportions, and assembly requirements.

MATERIALS: Expected service environment, required performance, formability, and cost are the principal factors influencing the specification of bellows materials, TABLE 2.

A brass alloy consisting of 80 per cent copper and 20 per cent zinc is most commonly used because of its suitability for the majority of applications. However, this alloy is not suitable under severely corrosive conditions nor if traces of mercury or ammonia are present. Under certain conditions, hot water will also shorten bellows life.

Brass bellows with a silver coating have been used for many years under severely corrosive conditions. Not a plating, the silver is solid, rolled on the surface of the brass to a thickness of 0.0006-inch, either inside or outside the bellows.

A bronze alloy of 95 per cent copper and 5 per cent tin is used where a slightly higher corrosion resistance than that of bare brass is required. Bronze bellows are also used in instrument type service where hysteresis may be critical. Heat treatment of bronze bellows reduces hysteresis to a negligible value for most practical applications.

Monel, consisting of approximately 40 per cent copper and 60 per cent nickel, is frequently used for its corrosion resistance properties. It has good salt spray resistance and serves well in contact with water. Monel is more difficult to draw than brass, and accordingly, is not available in extraflexible deeper convolutions.

Stainless steels are being increasingly used for bellows because of their excellent corrosion resistance and high-temperature characteristics. Some limitations in design proportions arise, by comparison with other materials mentioned, because of drawing and annealing problems. For extra high-temperature service bellows have also been fabricated from Inconel.

Since some mystery is often associated with hysteresis in springs or springlike members, the subject might well be clarified at this point. Hysteresis in bellows is defined according to the representation in *Fig.* 14. Although spring rate is the same in loading or unloading, the displacements or movements in the two directions between the same load or force limits are different. Hysteresis is expressed as the ratio of H to F.

Different materials and different types of bellows have different hysteresis values. Hysteresis can be reduced by special heat treating. For example, regular brass bellows show 4 per cent hysteresis but when heat treated, only 2 per cent. Regular bronze bellows have a hysteresis value of about 1 per cent, and when heat treated, 0.5 per cent. For the same materials and conditions, but with extraflexible bellows, hysteresis values are only about one-third the magnitude of those cited.

In a bellows device, hysteresis is sometimes thought to be responsible for lags in response greatly in excess of the foregoing data. However, friction in the device operated by the bellows is more often found to be the principal source of error.

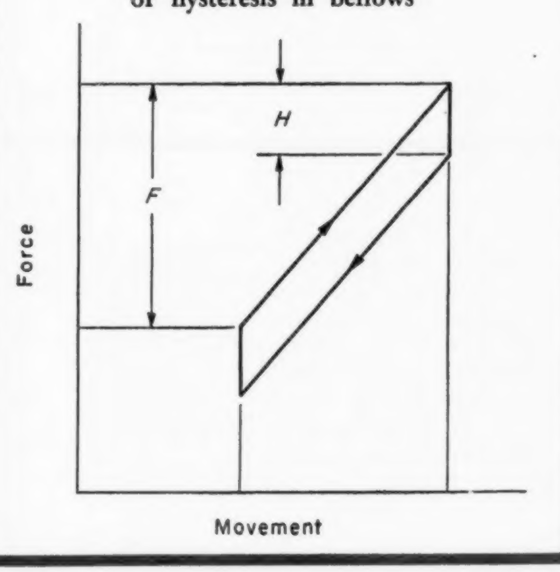
DESIGN FACTORS: The properties of a bellows are somewhat like those of a spring, but in design the number of variables are fewer and calculation is less involved. Generally speaking with respect to the bellows itself, the same principles hold regardless of the type of application.

As indicated before, two basic types of bellows are usually available — regular and extraflexible. The regular style bellows has a depth of convolution such that the outside diameter is approximately 50 per cent larger than the inside diameter. In the extraflexible bellows, the outside diameter is

Table 2—Common Bellows Materials

Material	Corrosion Resistance	Max. Temp. (F)	Ease of Fabrication	Cost
Brass	Fair	350	Excellent	1
Brass, silver clad	Excellent	350	Excellent	3
Bronze	Fair	350	Excellent	2
Monel	Good	900	Fair	4
Stainless Steel	Excellent	1100	Fair	5
Inconel	Excellent	1500	Fair	6

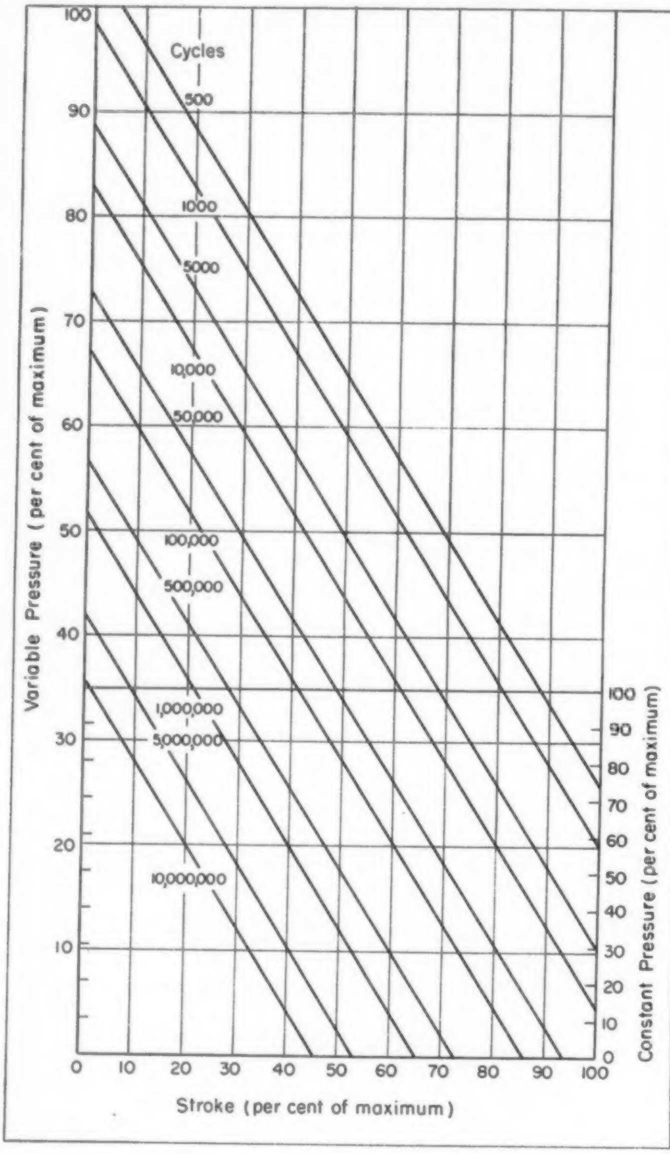
Fig. 14—Representation of hysteresis in bellows



approximately 60 per cent larger than the inside.

Besides the convolution depth, the factors that can be selected to fit a bellows to a given application are bellows size (diameter), wall thickness, and number of convolutions. Bellows travel and operating pressure are normally dictated by the application, of course. All these factors must be properly combined and balanced to guarantee adequate bellows life.

Fig. 15—Life expectancy chart for constant-pressure and variable-pressure bellows applications



Within the range of most common usage, 1/4 to 4 1/2-inch outside diameter, bellows manufacturers usually have a more or less standardized series of bellows that fill all needs likely to arise. Table 3 shows for only 1 1/8-inch diameter brass bellows the various engineering data on which bellows selection is based. All standard sizes in brass would total well over 100. Besides data contained in Table 3, other information required is given by the life expectancy chart in *Fig.* 15.

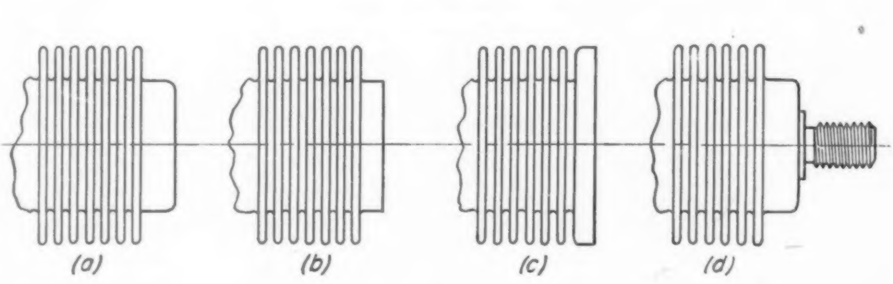
Selecting a bellows on the basis of required stroke and pressure is the normal procedure and is relatively easy with bellows data of the sort excerpted in Table 3. However, life expectancy can be a critical factor and may well influence the final choice. Therefore, the following example problem starts with a required life in order that the effect of this quality might be directly observed.

Assume an application requires a bellows with a service life of about 1,000,000 cycles operating in a system in which the pressure remains constant during the working stroke. From *Fig.* 15, the desired combination of constant pressure and stroke, both in terms of per cent of their maximums, must be selected. Assume that length of stroke is most important, that there is required a bellows which has a stroke of about 3/16-inch and is as small as possible in relation to its working stroke. Then, as high a per cent of maximum stroke as seems feasible, in keeping with the pressures involved, would be chosen.

Fig. 15 shows that if a bellows is selected that must deflect more than 66 per cent of its maximum stroke, the system cannot have any pressure on it —unless shorter life can be accepted. Assume that the system has pressure, but that it is relatively low. The stroke and pressure combination selected should reflect the relative importance of these two factors in design.

As a trial, there might be selected a bellows that will deflect 60 per cent of its maximum allowable stroke. On *Fig.* 15, intersection of 1,000,000 cycles and 60 per cent maximum stroke corresponds to 14 per cent maximum constant pressure. In other words the bellows selected from data such as given

Fig. 16—Common end designs of bellows: *(a)* closed end, *(b)* open end at root, *(c)* open end at major diameter, *(d)* attached standard fitting. Any pair of these designs can be used on a bellows



in TABLE 3 must have a pressure rating great enough that 14 per cent of it is equal to or greater than the anticipated system pressure.

Assume now that a bellows of about 1⅛-inch outside diameter would be acceptable for size. As TABLE 3 shows for regular 1⅛-inch bellows, four wall thicknesses are available. Since the pressure is low, the minimum wall thickness, 0.004-inch, might be tried so that greatest flexibility is obtained. For this bellows, maximum deflection per convolution is 0.029-inch. But since only 60 per cent of the maximum deflection is to be permitted, the usable deflection per convolution is 0.60 (0.029) = 0.017-inch.

Next a pressure check is made. From TABLE 3 the maximum internal pressure is 55 psi. Allowance for 14 per cent of maximum means that the system pressure may approach 0.14 (55) = 7.70 psi gage.

If that pressure is satisfactory, the number of required convolutions can be determined. The number is obtained by dividing the required stroke by the usable deflection per convolutions or (3/16)/0.017 = 11 convolutions. According to TABLE 3, the normal free length of each convolution is 0.084-inch or the total free length is 11 (0.084) = 0.924 = 15/16-inch approximately. Thus, all required factors have been determined. Similar calculations can be made from a number of different starting conditions, of course.

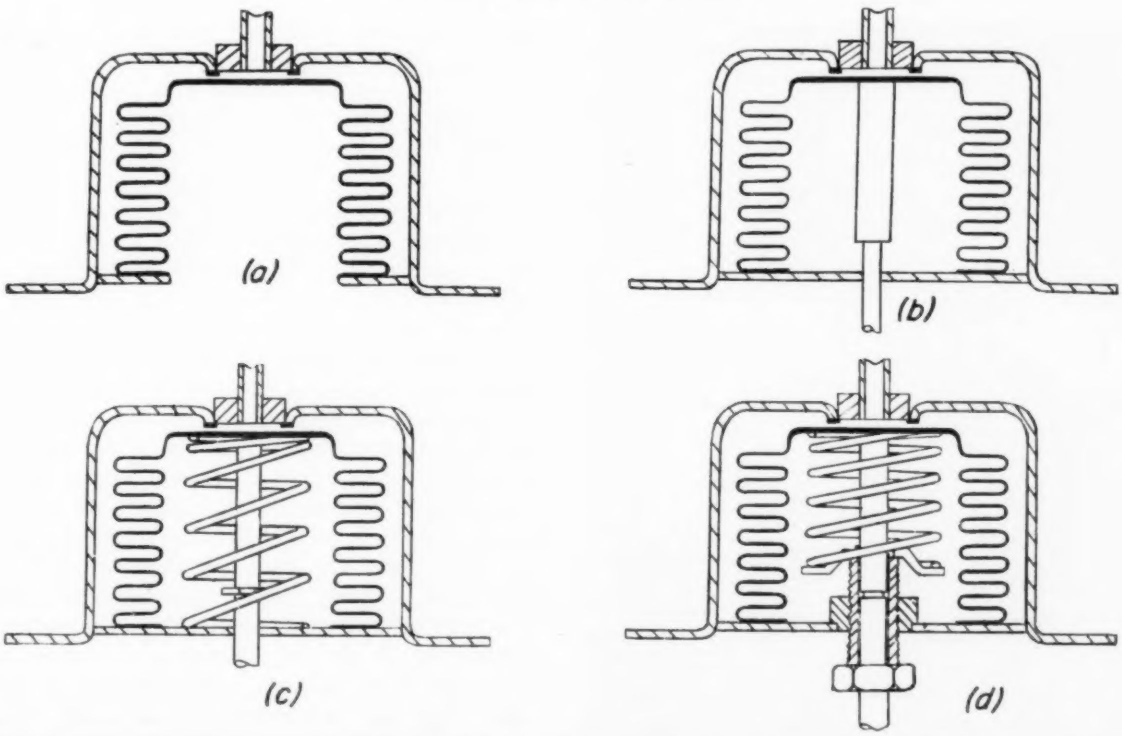
As apparent from the foregoing example as well as from *Fig.* 15, required service life is a major influence in bellows selection. Actually, in the outlined method, the life prediction is serving a purpose normally covered in design by two factors—the design stress and cyclical endurance at that stress. These two conventional criteria are lumped together in bellows engineering because no advantage is gained by their evaluation separately. Even this simplification leaves wide latitude for engi-

Table 3—Characteristics of 1⅛-inch Bellows

Characteristic	Regular				Extraflexible		
Root Diameter[1] (in.)	¾				21/32		
Convolutions[2] (standard)	11				11		
Effective Area[3] (sq in.)	0.69				0.62		
Wall Thickness (in.)	0.004	0.005	0.006	0.007	0.004	0.005	0.006
Approx. Free Length[4] in. per convolution)	0.084	0.079	0.074	0.070	0.100	0.087	0.076
Maximum Deflection[5]	0.029	0.028	0.027	0.023	0.047	0.034	0.029
Spring Rate[6] (lb per in. per convolution)	172	291	552	687	62	108	226
Flexibility[7] (in. per lb per sq in. per convolution)	0.0040	0.0024	0.0012	0.0010	0.010	0.006	0.003
Maximum Internal Pressure (psi)	55	80	150	200	55	65	130
Maximum External Pressure (psi)	60	88	165	220	61	72	143

[1] To obtain approximate inside diameter subtract 2 times wall thickness from root diameter.
[2] May be decreased in every case, increased in some.
[3] To obtain volume, multiply by bellows length.
[4] To obtain total free length of bellows, multiply by number of convolutions.
[5] To obtain total maximum deflection of bellows, multiply by number of convolutions.
[6] To obtain spring rate of bellows, divide by number of convolutions.
[7] To obtain flexibility of bellows, multiply by number of convolutions.

Fig. 17—Common types of bellows cup assemblies: *(a)* plain cup assembly, *(b)* extension-compression stop assembly, *(c)* spring-loaded assembly with stops, *(d)* adjustable spring-loaded assembly



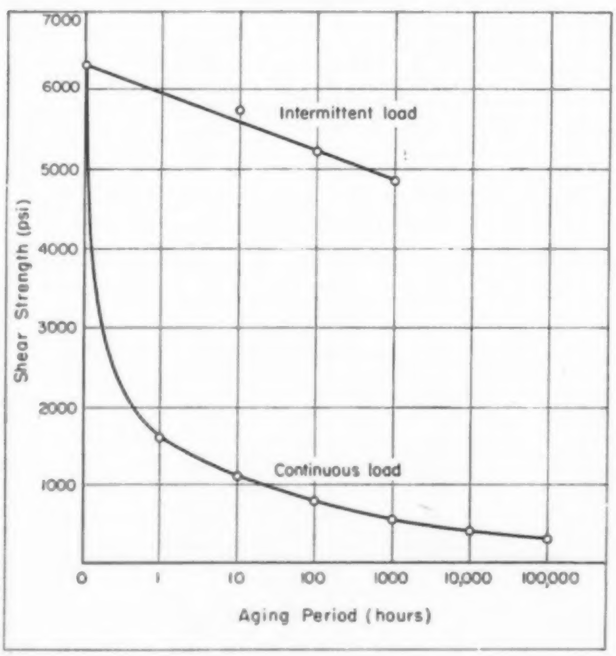
neering judgment, since the working stroke and pressure may have variable ranges in the application, operation may vary greatly in cyclical frequency, effects of environmental factors may vary widely, and so forth.

A distinction is made in *Fig.* 15 between variable pressure and constant pressure. For example, for 50 per cent of the maximum stroke, life is 500,000 cycles for 50 per cent of the maximum noted constant pressure, but only 5000 cycles for 50 per cent of maximum rated variable pressure. This difference indicates that even with no axial displacement of the bellows, changes in pressure cause internal movement of the bellows and definitely affect endurance.

A question frequently arises when an application requires a relatively high operating pressure and the pressure is variable around that high value. For example, the working pressure may be variable between 70 and 80 per cent of the rated bellows pressure for a 50 per cent maximum stroke. Should this condition be considered 80 per cent variable pressure (with practically no life expectancy), 80 per cent constant pressure (100,000 cycles), or in some other way?

Actually the unit is operating at 80 per cent constant pressure plus an additional factor, in terms of its effect upon life, of 10 per cent variable pressure. The chart gives 2,000,000 cycles for 10 per cent variable pressure. Life less than 100,000 cycles obviously should be predicted, but how much less becomes a matter of some judgment. If life is critical, the best solution is usually to alter the bellows proportions to allow a margin of safety. For example, if the foregoing conditions had applied for a 10-convolution bellows, convolutions might be increased to 11, a 10 per cent increase in free length, and stroke decreases from 50 to 45 per cent. Life expectancy for 80 per cent maximum constant pressure increases from 100,000 to more than 200,000, leaving an adequate margin for the 10 per cent variable pressure.

Fig. 18—Comparison of shear strength versus aging period of soldered (40 per cent tin, 60 per cent lead) brass joint under continuous and intermittent loads



A similar problem arises with respect to stroke. In a modulating control device, for example, the bellows may be called upon to operate from zero to 70, 80 or even 90 per cent of its maximum allowable stroke. However, the normal modulating movement occurring most of the time may be only 5 per cent of the total stroke. The major factor in such a case is the 5 per cent stroke. But again, however, some allowance must be provided for the occasional high-percentage stroke.

Proportions and properties of bellows can be summarized by a few general rules that may serve to guide both preliminary rough estimates and final calculations:

1. Flexibility, inches of stroke per psi of pressure, varies directly with the number of convolutions, directly with the square of the outside diameter, inversely with approximately the cube of wall thickness, inversely with the modulus of elasticity.
2. Spring rate, pounds of force per inch of movement, varies inversely with the number of convolutions, inversely with the square of the outside diameter, directly with approximately the cube of the wall thickness, directly with the modulus of elasticity.

These proportionalities also suggest the influences of manufacturing variations upon the accuracy of bellows performance. For example, a tolerance of ± 0.00025-inch on a wall thickness of 0.005-inch yields a range on flexibility of -16 per cent to $+13$ per cent, and other factors augment the arithmetic total to about ± 20 per cent. This maximum range, fortunately, is moderated by probabilities and in many designs, provision of adjustment or calibration means minimizes or nullifies differences from bellows to bellows.

Bellows Assemblies: For adaptation to the host of applications suggested in the foregoing sections, bellows are supplied by manufacturers at a number of assembly levels. These levels range progressively from the bellows alone (*Fig. 16a, b,* and *c*), bellows with end fittings (*Fig. 16d*), bellows in cup assemblies (*Fig.* 17) and complete units such as bellows, capillary and bulb for thermostats, thermal and pressure switches, etc.

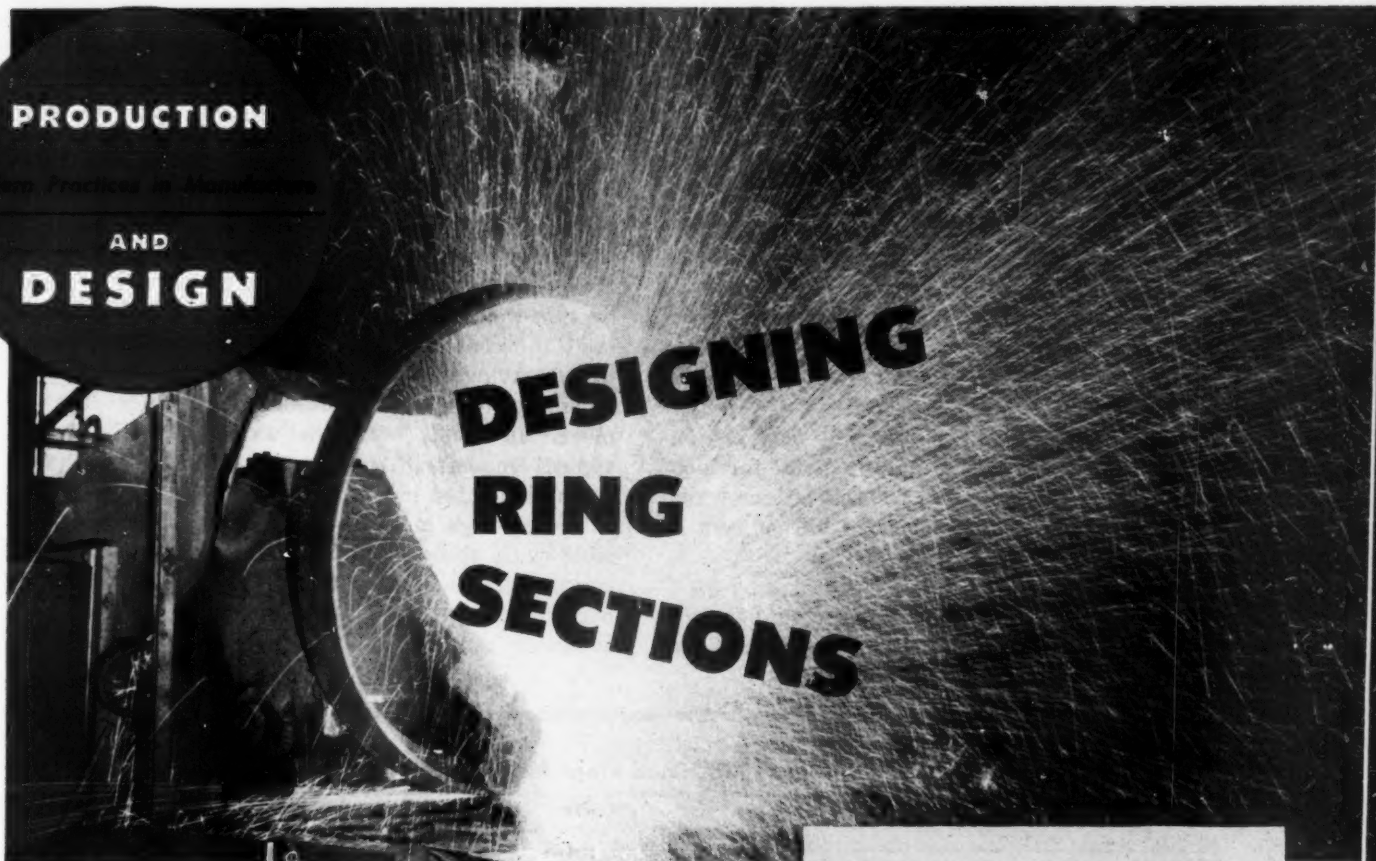
The physical joining of bellows to end fittings or other components is frequently a rather critical operation. Soldering is quite common for brass and bronze bellows but a number of precautions must be kept in mind. Fluxes containing ammonia or other corrosive agents cannot be used without impairing bellows life. Also aging of soldered joints reduces their strength, *Fig.* 18, and must be considered in design. Brazing and welding give excellent joint strength. Frequently no mechanical joint, such as formed by spinning, is needed with these methods.

PRODUCTION

Modern Practices in Manufacture

AND

DESIGN



DESIGNING RING SECTIONS

Use of flash-welded rolled sections offers considerable savings in materials and production time

By Verne Wildman

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CRITICAL or intricate circular parts of relatively delicate section often involve production and cost factors which must be considered in design. Manufacturing costs may often prove unfavorable in cases where rings of large diameter are required with a complex cross section. This is especially true where the amount of material to be machined away is excessive or the material is low in machinability.

Although simple ring-shaped parts can be produced by a number of methods, flash welding from rolled stock sections has proved to be economical, *Fig.* 1. Wide availability of flash-weldable materials and elimination of machining time by use of plain sheet or plate stock or specially rolled sections offers attractive cost advantages.

Fig. 1—Top—Flash-welding a formed rolled section into a one-piece homogeneous ring

Fig. 2—Right—Two completed simple flash-welded rings showing wide variation in size and section possible

Size Ranges: In general, flash-welded rings range from a minimum diameter of around 5 inches to a maximum diameter of 8 feet. Thickness of section possible covers the general range available in rolled plate, or bar, *Fig.* 2. Width of section which normally can be handled may range to about a maximum of 18 inches.

Materials and Properties: Few restrictions in materials selection are present. Metals which have been handled successfully in production range from ingot iron to bearing steels. Most of the stainless alloys are suitable and aluminum and titanium rings have been produced in quantity.

Properties of the flash-welded joint may be considered as equivalent to the parent metal for most purposes. All welds meet AAF specifications which require a minimum for all welds of 95 per cent of parent metal properties.

Special Mill Sections: While appreciable savings are possible with parts produced from square and rectangular sections, very often special mill-rolled sections can be used to even greater advantage. Some parts, such as the tank bogie wheel shown in *Fig.* 3, can be produced to final size which eliminates practically all machining. Where final machining is necessary, and especially in the relatively more expensive alloys, special mill sections are equally advantageous, *Fig.* 4. Based only on the added cost of a special rolled section, wherever there is a saving of approximately 25 per cent or over in weight (compared to the usual rectangular bar) the cost savings are well worthwhile. Additional savings in machining result in a total cost advantage frequently running to 50 per cent or more. In TABLE 1 the cost savings are shown for typical machined rings made from special sections illustrated in *Fig.* 5.

The savings in critical materials which are possible with the use of mill sections are of major concern in time of war, and have on many such occasions more than justified going beyond purely economic considerations in favoring their use. Or-

Table 1—Cost Savings on Machined Rings Made from Special Sections

Ring	Description	Weight (lb)	Cost Savings (per cent) Rough Ring	Machining	Finished Ring
Type 347	Rectangular Bar $1\frac{1}{32}$ x $1\frac{5}{8}$	33	...	...	...
Stainless	Mill Section *a**	22.1	10.2	29.3	16.7
Type 321	Rectangular Bar $1\frac{3}{16}$ x $2\frac{5}{16}$	42.4	...	...	...
Stainless	Mill Section *b**	27.7	26.1	8.3	17.0
Type 321	Rectangular Bar $1\frac{31}{32}$ x $5\frac{5}{16}$ (two pieces)	266	...	...	...
Stainless	Mill Section *c** (two pieces)	146.6	43.8	16.5	28.6
Type 347	Rectangular Bar $1\frac{1}{16}$ x $2\frac{1}{32}$	68	...	...	...
Stainless	Mill Section *d**	44¼	32.5	41.4	35.8

* Details shown in *Fig.* 5.

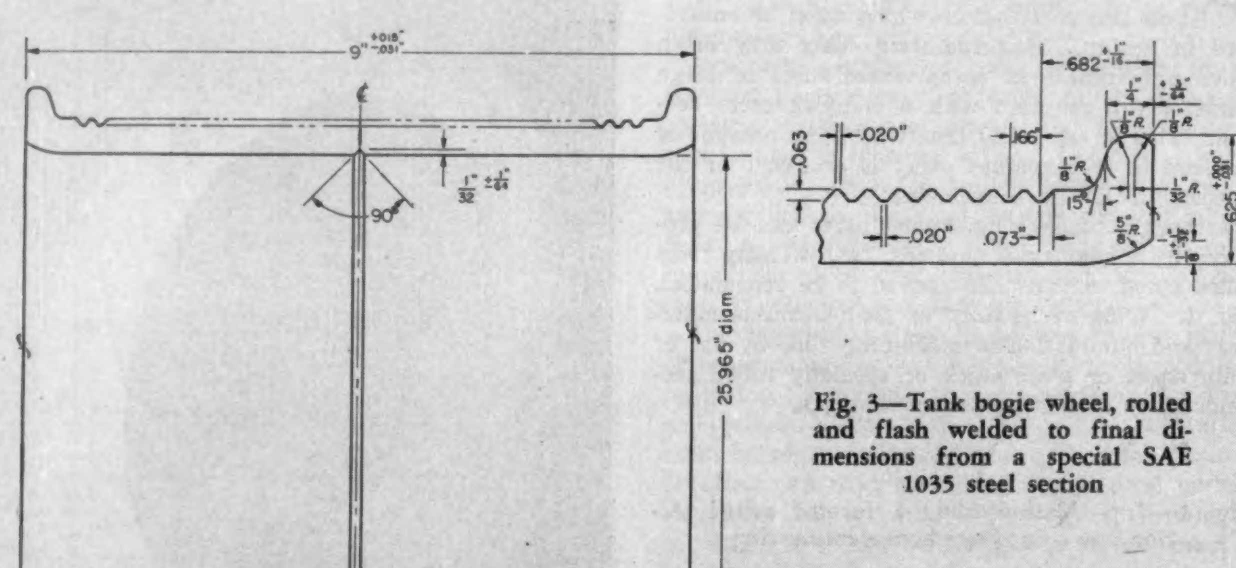


Fig. 3—Tank bogie wheel, rolled and flash welded to final dimensions from a special SAE 1035 steel section

dinarily, however, some consideration of production quantities must be made to evaluate and establish the suitability of a design for the use of special sections. The quantity of rings, multiplied by the saving per ring must at least equal the cost of the special rolls required to produce the mill sections for the practice to be economically justified. And it must not be overlooked that the mills themselves have been known to impose minimum quantities they will roll.

Thus, in addition to the decrease in raw material weight mentioned, a sufficient quantity of special stock must be needed to justify and make practical the rolling process. A minimum quantity for rolling could be considered as 10,000 pounds, but material and other factors may adjust this figure upwards. Generally, 20,000 pounds is considered an average for suitable economy. However, on large sections this ranges from 10 to 25 tons. The necessity for suitable rolling tonnage is made evident from the fact that special roll sets cost from $4000 to $7000.

Design Factors: In addition to the several considerations for design already indicated, there are some special possibilities that should be recognized. These concern parts in which the circular uniform section constitutes a major portion of the unit, but not all. Such parts, having only small nonuniform

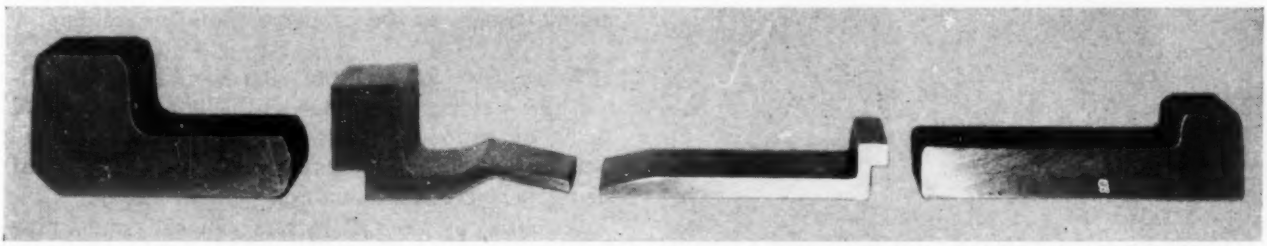
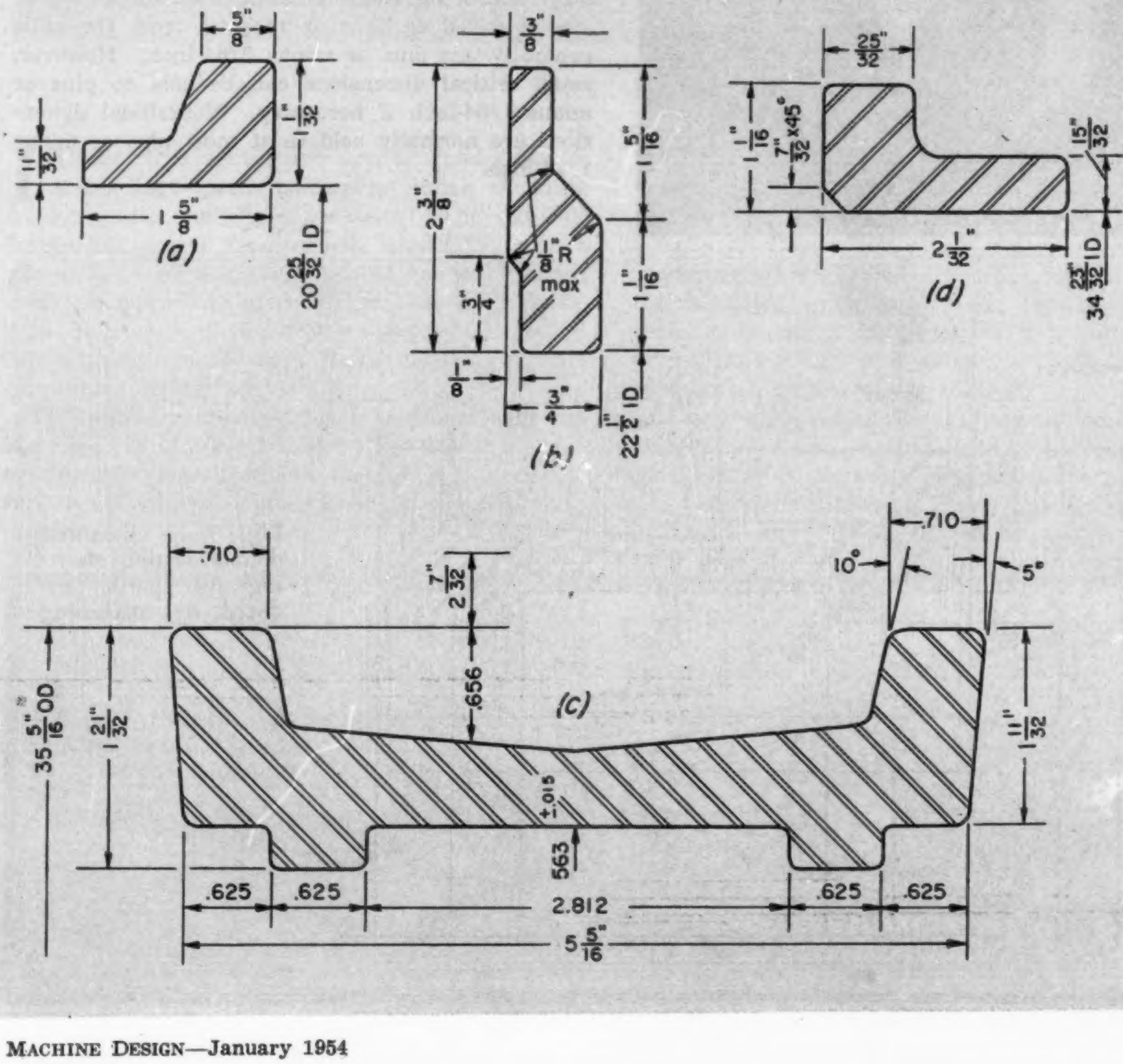


Fig. 4—Above—Group of typical special rolled sections utilized for rolled and flash-welded rings

Fig. 5—Below—Details of mill sections, the per cent saving for which are shown in Table 1



or specially contoured projections, can often be designed for production by this method with similar advantage. The ring shown in *Fig.* 6 is a good illustration of this type of part.

On parts formed from special rolled sections, corner radii range from 1/16 to 3/16-inch, ⅛-inch being an average corner and fillet radius used. Large radii, however, simplify the rolling operation.

For machining operations on special rolled sections, stock allowance of 5/64 to 3/32-inch is generally employed. Allowance on inside and outside diameters over finished dimensions may range up to ¼-inch on the diameter, depending on size and other conditions.

Where greatly unbalanced or asymmetrical sections, which are difficult to roll, are necessary it is sometimes possible to utilize a combination section. By combining two sections into one rolled cross-section, a balanced design results. Such a section is shown in *Fig.* 7. Here, two separate but similar parts are produced from the one welded ring with considerable advantage. In other cases a protruding leg of a section, easiest to roll straight, might be bent to the proper angle after welding.

Tolerances: Accuracy of ring sections after rolling and welding is obtained by a final hot or cold sizing operation. Rolling for welding includes the preparation of suitable flats for flash welding. Final sizing perfects the roundness and accuracy.

Variation of sized diameters generally runs plus or minus 1/16-inch. These tolerances as generally expressed include not only variations of actual from nominal diameters but any out-of-roundness which may exist in the ring. Tolerances on dimensions of special rolled sections as received from the mills generally are plus or minus 3/64-inch. However, small critical dimensions can be held to plus or minus 1/64-inch if necessary. Noncritical dimensions are normally held to at most plus or minus 1/16-inch.

Fig. 6—Ring part with welded in section having projecting ears

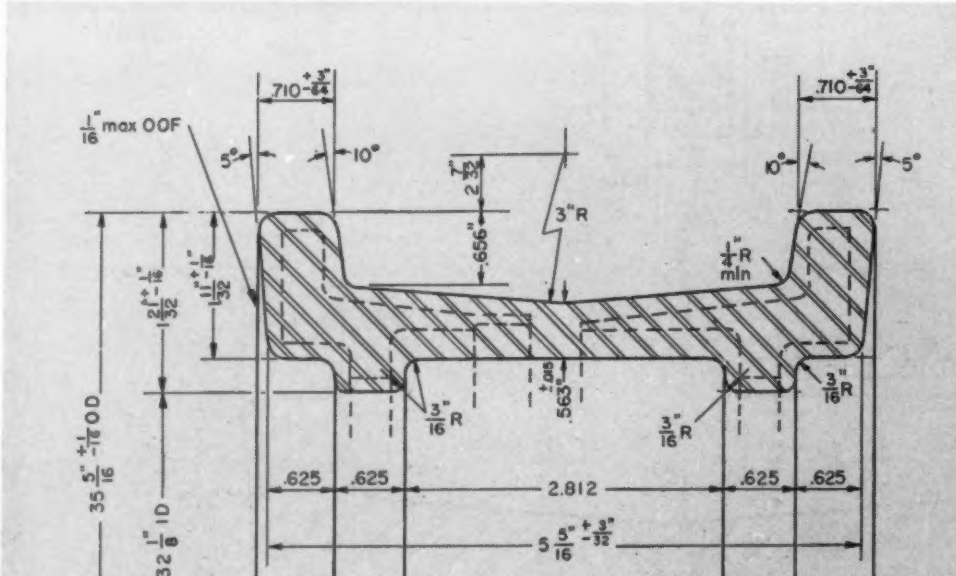


Fig. 7 — Balanced double section showing two parts produced by machining

KINEMATIC ANALYSIS

... a method for determining characteristics of such mechanisms as

- *oscillating beam*
- *crank and connecting rod*
- *four-bar linkage*

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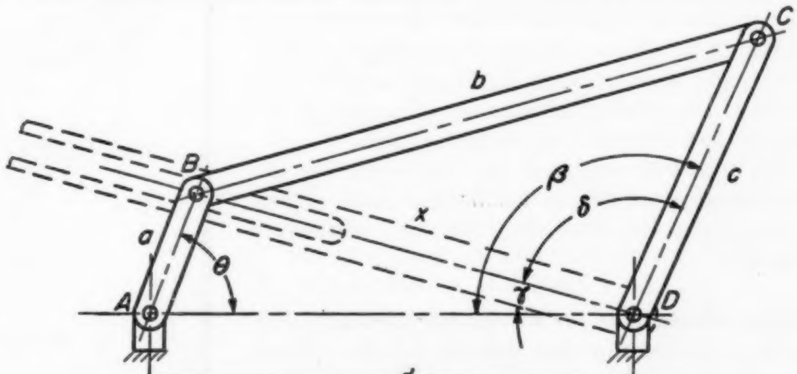
INCREASING speeds and cycling rates in automatic high-speed machinery make accurate kinematic analysis a necessity. The dynamic forces and power requirements associated with a given mechanism cannot be determined without complete knowledge of its kinematics. It is therefore the purpose of this article to present a method of analysis of mechanisms that will fulfill any accuracy requirements.

The approach in this analysis is general and can be used for any mechanism. The result of this approach provides a scheme that in most cases will render the kinematic analysis less time-consuming than the graphical methods. The tabulated form presented may be used without recourse to the mathematics involved in deriving the kinematic equations. Consequently, a working knowledge of trigonometry is the only requisite for performing the indicated calculations. On the other hand, those interested in the proof will find that the approach used in this analysis can be applied to more complicated mechanisms.

In this method the general formulations for the kinematics of any machine are first developed in the form of useful relationships between the absolute and relative motions. These general equations

Fig. 1—Four-bar linkage, with oscillating beam shown phantomed



are then applicable to the analysis of particular mechanisms. Simple mechanisms are easily treated and more complex ones usually are most conveniently analyzed as combinations of the simple ones. For example, the four-bar linkage is a combination of the oscillating beam and the crank and connecting rod, and its analysis is conducted on this basis. The actual tabulation scheme will be presented and discussed in relation to this analysis.

General Formulation: A mechanism in general is characterized by the kinematic relationship between the output to input. The output characteristics stem mainly from two sources: the relative kinematics of the system and the absolute kinematics of the input. Relative kinematics is defined as the relation between output to input, and is determined by the geometry of the mechanism.

The output velocity of a mechanism is given by

$$\frac{d\beta}{dt} = \frac{d\beta}{d\theta}\frac{d\theta}{dt} \quad \text{(1)}$$

where $d\beta/d\theta$ is the ratio of angular velocities, and represents the effect of the relative kinematics of the mechanism on the output velocity.

The angular acceleration of the output is obtained by differentiating Equation 1 with respect to time t:

$$\frac{d^2\beta}{dt^2} = \frac{d}{dt}\left(\frac{d\beta}{dt}\right) = \frac{d}{dt}\left(\frac{d\beta}{d\theta}\frac{d\theta}{dt}\right)$$

$$= \frac{d\beta}{d\theta}\frac{d^2\theta}{dt^2} + \frac{d\theta}{dt}\frac{d}{dt}\left(\frac{d\beta}{d\theta}\right)$$

but

$$\frac{d}{dt}\left(\frac{d\beta}{d\theta}\right) = \frac{d}{d\theta}\left(\frac{d\beta}{d\theta}\right)\frac{d\theta}{dt} = \frac{d^2\beta}{d\theta^2}\frac{d\theta}{dt}$$

Therefore

$$\frac{d^2\beta}{dt^2} = \frac{d\beta}{d\theta}\frac{d^2\theta}{dt^2} + \left(\frac{d\theta}{dt}\right)^2\frac{d^2\beta}{d\theta^2} \quad \text{(2)}$$

where $d\beta/d\theta$ and $d^2\beta/d\theta^2$ are the terms representing the effect of the relative kinematics on the acceleration of the output. For constant angular velocity of the input $d^2\theta/dt^2 = 0$ and

$$\frac{d^2\beta}{dt^2} = \left(\frac{d\theta}{dt}\right)^2\frac{d^2\beta}{d\theta^2} \quad \text{(2a)}$$

Examination of Equations 1 and 2 shows that for given inputs $d\theta/dt$ and $d^2\theta/dt^2$ the absolute output velocities and accelerations $d\beta/dt$ and $d^2\beta/dt^2$ can be found if the relative kinematics of output to input $d\beta/d\theta$ and $d^2\beta/d\theta^2$ can be formulated.

In the following sections the equations for the relative kinematics of the oscillating beam, crank and connecting rod, and the four-bar linkage are developed.

Relative Kinematics for a Four-Bar Linkage: *Fig.* 1 shows a typical four-bar linkage *ABCD*. In addition, a slotted link x is shown pivoted freely about *D*. It can be seen that the slotted link does not affect the positioning of the four-bar linkage. Hence, if the input arm a is turned through an angle θ, the slotted link merely orients itself to conform to the independent positioning of the four-bar linkage.

The addition of the slotted link is shown dotted to emphasize that it is extraneous to the basic mechanism. The figure graphically bears out the fact that the kinematics of a four-bar linkage is simply the algebraic sum of the kinematics of an oscillating beam *ABD* and a crank and connecting rod *DBC*.

A rigorous proof of the foregoing statement follows. From the geometry of *Fig.* 1,

$$\beta = \gamma + \delta \quad \text{(3)}$$

Differentiating Equation 3 with respect to time t yields the angular velocity relation

$$\frac{d\beta}{dt} = \frac{d\gamma}{dt} + \frac{d\delta}{dt} \quad \text{(4)}$$

Differentiating Equation 4 with respect to time t yields the angular acceleration relationship

$$\frac{d^2\beta}{dt^2} = \frac{d^2\gamma}{dt^2} + \frac{d^2\delta}{dt^2} \quad \text{(5)}$$

The mathematics of Equations 4 and 5 indicate that γ and δ are independent variables. However, in the four-bar linkage γ and δ are parametrically related by the angle θ.

Differentiating Equation 3 with respect to θ gives

$$\frac{d\beta}{d\theta} = \frac{d\gamma}{d\theta} + \frac{d\delta}{d\theta} \quad \text{(4a)}$$

and differentiating again with respect to θ gives

$$\frac{d^2\beta}{d\theta^2} = \frac{d^2\gamma}{d\theta^2} + \frac{d^2\delta}{d\theta^2} \quad \text{(5a)}$$

Substituting Equations 4*a* and 5*a* into Equations 1 and 2, respectively, produces

$$\frac{d\beta}{dt} = \left(\frac{d\gamma}{d\theta} + \frac{d\delta}{d\theta}\right)\frac{d\theta}{dt} \quad \text{(6)}$$

and

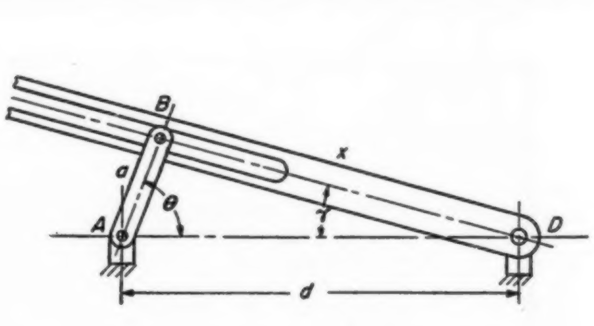


Fig. 2—Oscillating beam

$$\frac{d^2\beta}{dt^2} = \left(\frac{d\gamma}{d\theta} + \frac{d\delta}{d\theta}\right)\frac{d^2\theta}{dt^2} + \left(\frac{d\theta}{dt}\right)^2\left(\frac{d^2\gamma}{d\theta^2} + \frac{d^2\delta}{d\theta^2}\right) \quad \ldots\ldots(7)$$

thus rendering the output velocity and acceleration in terms of γ and δ. Therefore, in order to find the output velocity $d\beta/dt$ and the output acceleration $d^2\beta/dt^2$ it is necessary to determine $d\gamma/d\theta$, $d\delta/d\theta$, $d^2\gamma/d\theta^2$ and $d^2\delta/d\theta^2$.

The purpose of establishing the velocity and acceleration Equations 6 and 7 in terms of γ and δ for the four-bar linkage is as follows: γ is applicable as a variable for the oscillating beam, while δ is applicable as a variable for the crank and connecting rod. Thus the equations for the four-bar linkage show, mathematically, their relation to the two simpler mechanisms.

Oscillating Beam Mechanism: In *Fig.* 2 the oscillating beam mechanism *ABD* from *Fig.* 1 has been extracted and reproduced for its analysis. It will be shown later that the equations derived for the oscillating beam apply also to both the internal and external Geneva mechanisms.

From the geometry of *Fig.* 2, the following relationship is established:

$$\sin\gamma = \frac{a\sin\theta}{x} \quad \ldots\ldots(8)$$

where, by the law of cosines,

$$x = \sqrt{a^2 + d^2 - 2\,ad\cos\theta} \quad \ldots\ldots(9)$$

Thus

$$\gamma = \sin^{-1}\frac{a\sin\theta}{x} \quad \ldots\ldots(10)$$

Equation 10 is the angular displacement of the oscillating beam.

Now, differentiating Equation 10 with respect to θ and arranging terms, the relative velocity of the output to input is

$$\frac{d\gamma}{d\theta} = \frac{ad\cos\theta - a^2}{a^2 + d^2 - 2\,ad\cos\theta} = \frac{ad\cos\theta - a^2}{x^2} \quad \ldots\ldots(11)$$

Differentiating Equation 11 with respect to θ,

$$\frac{d^2\gamma}{d\theta^2} = \frac{(a^2 - d^2)\,ad\sin\theta}{(a^2 + d^2 - 2\,ad\cos\theta)^2} = \frac{(a^2 - d^2)\,ad\sin\theta}{x^4} \quad \ldots\ldots(12)$$

Equations 11 and 12 when substituted into Equations 1 and 2 with γ replacing β, provide the absolute velocities and accelerations of the oscillating beam.

Equations 10, 11 and 12 give the complete kinematics of any oscillating beam or quick-return mechanism. These equations are useful in design in that one can find the accelerations and velocities of the beam and hence the loads, stresses, and power requirements necessary for operation.

Nomenclature

θ =	Input angle
β =	Output angle
$d\theta/dt$ =	Input angular velocity
$d\beta/dt$ =	Output angular velocity
$d^2\theta/dt^2$ =	Input angular acceleration
$d^2\beta/dt^2$ =	Output angular acceleration

Crank and Connecting Rod Mechanism: In *Fig.* 3, the crank and connecting rod mechanism *DCB* from *Fig.* 1 was extracted and reproduced for analysis. In this analysis, the slotted link is held fixed, so that a change in δ relative to a change in x can be formulated. From *Fig.* 3, and by the law of cosines,

$$b^2 = x^2 + c^2 - 2xc\cos\delta \quad \ldots\ldots(13)$$

Solving for x in terms of δ, the displacement relationship for the crank and connecting rod mechanism is found:

$$x = c\cos\delta + \sqrt{c^2\cos^2\delta - (c^2 - b^2)} \quad \ldots\ldots(14)$$

Differentiating Equation 14 with respect to δ gives the velocity relationship for the crank and connecting rod:

$$\frac{dx}{d\delta} = -c\sin\delta\left(1 + \frac{c\cos\delta}{\sqrt{c^2\cos^2\delta - (c^2 - b^2)}}\right) \quad (15)$$

Another form for the velocity relation is derived by differentiation of Equation 13:

$$\frac{dx}{d\delta} = \frac{xc\sin\delta}{c\cos\delta - x} \quad \ldots\ldots(16)$$

Differentiating Equation 16 results in the accelera-

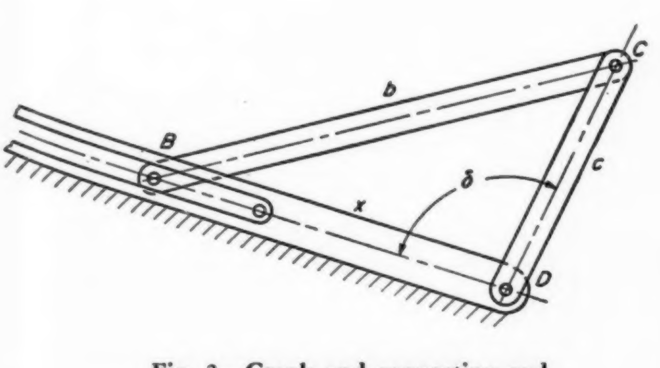


Fig. 3—Crank and connecting rod

Col. No.	(1)	(2)	(3)	(4)	(5)	(6)	(7)
Function	θ	$\sin\theta$	$\cos\theta$	$a \sin\theta$	$2ad \cos\theta$	x^2	x
Oper	Oper. Angle	sin (1)	cos (1)	$\left.\frac{a}{.9235}\right\}$x(2)	$\left.\frac{2ad}{9.3772}\right\}$x(3)	$\left.\frac{a^2+d^2}{26.6288}\right\}$-(5)	$\sqrt{(6)}$
1	0°	0	1.0000	0	9.3772	17.2516	4.1535
2	10°	.1736	.9848	.1604	9.2348	16.3940	4.1706
9	80°	.9848	.1736	.9095	1.6284	25.0004	5.0000
10	90°	1.0000	0	.9235	0	26.6288	5.1603
11	100°			.9095	-1.6284	28.2571	5.3157
18	170°			.1604	-9.2348	35.8636	5.9886
19	180°			0	-9.3772	36.0060	6.0005
20	190°						
36	350°						
37	360°						

Col. No.	(15)	(16)	(17)	(18)	(19)	(20)	(21)
Function	$\frac{d^2\gamma}{d\theta^2}$	$2cx$	$c^2-b^2+x^2$	$\cos\delta$	δ	$\sin\delta$	$c\sin\delta$
Oper.	$\frac{(13)}{(14)}$	$\left.\frac{2c}{4.00}\right\}$x(7)	$\left.\frac{c^2-b^2}{-18.6195}\right\}$+(6)	$\frac{(17)}{(16)}$	$\cos^{-1}$(18)	sin (19)	$\left.\frac{c}{2.0000}\right\}$x(20)
1	0	16.614	-1.3680	-.0823	94° 43′	.9960	1.9920
2	-.0671	16.682	-1.2255	-.0735	94° 13′	.9973	1.9946
9	-.1841	20.000	6.3809	.3190	71° 24′	.9478	1.8956
10	-.1648	20.641	8.0092	.3880	67° 10′	.9217	1.8434
11	-.1441	21.263	9.6376	.4533	63° 03′	.8914	1.7828
18	-.0158	23.954	17.2440	.7199	43° 57′	.6941	1.3882
19	0	24.002	17.3865	.7244	43° 35′	.6894	1.3788
20	.0158				43° 57′		
36	.0671				94° 13′		
37	0				94° 43′		

Col. No.	(29)	(30)	(31)	(32)	(33)	(34)	(35)
Function	$\frac{d\beta}{d\theta}$	$\left(\frac{dx}{d\theta}\right)^2$	$\frac{3}{x}$	$\frac{3}{x} \times \frac{d\delta}{dx}$	$\frac{1}{xc\sin\delta}$	$\cot\delta$	$\left(\frac{d\delta}{dx}\right)^2$
Oper.	(12) + (28)	$(27)^2$	$\frac{3}{7}$	(31) x (25)	$\frac{1}{(23)}$	cot (19)	$(25)^2$
1	.2223	0	.7223	-.3771	.1209	-0.0825	.2724
2	.1151	.0381	.7193	-.3732	.1202	-0.0737	.2694
9	-.4266	.8529	.6000	-.2760	.1055	0.3365	.2118
10	-.4508	.8256	.5814	-.2679	.1051	0.4211	.2124
11	-.4632	.7545	.5644	-.2625	.1055	0.5084	.2165
18	-.2269	.0185	.5010	-.2742	.1203	1.0373	.2994
19	-.1539	0	.5000	-.2751	.1209	1.0507	.3027
20	-.0781						
36	.3177						
37	.2223						

Fig. 4—Tabulating scheme for calculating kinematics of four-bar linkage

Col. No.	(8)	(9)	(10)	(11)	(12)	(13)	(14)
Function	$\sin\gamma$	γ	$ad\cos\theta$	$ad\cos\theta - a^2$	$\frac{d\gamma}{d\theta}$	$(a^2-d^2)ad\sin\theta$	x^4
Oper.	$\frac{(4)}{(7)}$	$\sin^{-1}(8)$	$\left.\frac{ad}{4.6886}\right\}\times(3)$	$(10)-\left\{\frac{a^2}{.8528}\right.$	$\frac{(11)}{(6)}$	$\left.\frac{(a^2-d^2)ad}{-116.85}\right\}\times(2)$	$(6)^2$
1	0	0	4.6886	3.8358	.2233	0	297.62
2	.0385	2° 12′	4.6174	3 7645	.2164	− 20.30	302.55
9	.1819	10°29′	0.8142	−0.0387	−.0016	−115.08	625.00
10	.1790	10°19′	0	−0.8529	−.0320	−116.85	709.10
11	.1711	9°51′	−0.8142	−1.6670	−.0590	−115.08	798.46
18	.0268	1°32′	−4.6174	−5.4702	−.1525	− 20.30	1286.23
19	0	0	−4.6886	−5.5415	−.1539	0	1296.43
20		−1° 32′			−.1525		
36		−2°12′			.2164		
37		0			.2223		

Col. No.	(22)	(23)	(24)	(25)	(26)	(27)	(28)
Function	$c\cos\delta$	$xc\sin\delta$	$c\cos\delta - x$	$\frac{d\delta}{dx}$	$ad\sin\theta$	$\frac{dx}{d\theta}$	$\frac{d\delta}{d\theta}$
Oper.	$\left.\frac{c}{20000}\right\}\times(18)$	$(7)\times(21)$	$(22)-(7)$	$\frac{(24)}{(23)}$	$\left.\frac{ad}{4.6886}\right\}\times(2)$	$\frac{(26)}{(7)}$	$(25)\times(27)$
1	−0.1646	8.2738	−4.3182	−.5219	0	0	0
2	−0.1470	8.3187	−4.3175	−.5190	0.8142	.1952	−.1013
9	0.6380	9.4780	−4.3619	−.4602	4.6174	.9235	−.4250
10	0.7760	9.5120	−4.3843	−.4609	4.6886	.9086	−.4188
11	0.9066	9.4763	−4.4092	−.4653	4.6174	.8686	−.4042
18	1.4398	8.3134	−4.5489	−.5472	0.8142	.1360	−.0744
19	1.4488	8.2735	−4.5517	−.5502	0	0	0
20							+.0744
36							.1013
37							0

Col. No.	(36)	(37)	(38)	(39)	(40)	(41)	(42)
Function	$\cot\delta\left(\frac{d\delta}{dx}\right)^2$			$\frac{ad\cos\theta}{x}$	$\frac{d\delta}{dx}\ \frac{ad\cos\theta}{x}$	$\frac{d^2\beta}{d\theta}$	β
Oper.	$(34)+(35)$	$-[(32)+(33)+(36)]$	$(30)\times(37)$	$\frac{(10)}{(7)}$	$(25)\times(39)$	$(40)+(38)+(15)$	$(9)+(19)$
1	−.0225	.2787	0	1.1288	−.5891	−.5891	94°43′
2	−.0199	.2729	.0104	1.1071	−.5746	−.6313	96° 25′
9	.0713	.0992	.0846	.1628	−.0749	−.1744	81° 52′
10	.0894	.0734	.0606	0	0	−.1042	77° 28′
11	.1007	.0563	.0425	−.1532	.0713	−.0303	72° 54′
18	.3106	−.1567	−.0029	−.7710	.4219	.4032	45° 29′
19	.3180	−.1638	0	−.7814	.4299	.4299	43° 35′
20			−.0029		.4219	.4348	42° 25′
36			.0104		−.5746	−.4971	92° 01′
37			0		−.5891	−.5891	94° 43′

tion relationship for the crank and connecting rod:

$$\frac{d^2x}{d\delta^2} = \left(\frac{1}{c\,(\cos\delta) - x} + \frac{2}{x} \right) \left(\frac{dx}{d\delta} \right)^2 + \frac{xc\cos\delta}{c\,(\cos\delta) - x} \quad \ldots\ldots (17)$$

where x is given by Equation 14. Equations 14 through 17 give the complete kinematics for a crank and connecting rod mechanism.

Four-Bar Linkage: The analysis of the four-bar linkage can now be presented using the results of the preceding analyses.

The displacement equation of the output crank c, *Fig.* 1, is

$$\beta = \gamma + \delta \quad \ldots\ldots (18)$$

where γ and δ are given by Equations 10 and 13. Therefore,

$$\beta = \sin^{-1}\frac{a\sin\theta}{x} + \cos^{-1}\frac{c^2 + x^2 - b^2}{2\,cx} \quad \ldots\ldots (19)$$

where x is defined by Equation 9. The basic relative velocity equation for the four-bar linkage is given in Equation 4*a* as

$$\frac{d\beta}{d\theta} = \frac{d\gamma}{d\theta} + \frac{d\delta}{d\theta}$$

where $d\gamma/d\theta$ is given in Equation 11 and $d\delta/d\theta$ can be written as

$$\frac{d\delta}{d\theta} = \frac{d\delta}{dx}\,\frac{dx}{d\theta}$$

Term $d\delta/dx$ is the reciprocal of Equation 16. In order to find $dx/d\theta$, Equation 9 is differentiated with respect to θ:

$$\frac{dx}{d\theta} = \frac{ad\sin\theta}{x} \quad \ldots\ldots (20)$$

The velocity ratio $d\beta/d\theta$ can be written as

$$\frac{d\beta}{d\theta} = \frac{d\gamma}{d\theta} + \frac{d\delta}{dx}\,\frac{dx}{d\theta} \quad \ldots\ldots (21)$$

The acceleration equation for the four-bar linkage is given in Equation 5*a* as

$$\frac{d^2\beta}{d\theta^2} = \frac{d^2\gamma}{d\theta^2} + \frac{d^2\delta}{d\theta^2}$$

where $d^2\gamma/d\theta^2$ is given in Equation 12 and $d^2\delta/d\theta^2$ is obtained by differentiating $d\delta/d\theta$ with respect to θ, as follows:

$$\frac{d^2\delta}{d\theta^2} = \frac{d}{d\theta}\left(\frac{d\delta}{d\theta}\right) = \frac{d}{d\theta}\left(\frac{d\delta}{dx}\,\frac{dx}{d\theta}\right)$$

$$= \left(\frac{dx}{d\theta}\right)^2 \frac{d^2\delta}{dx^2} + \frac{d\delta}{dx}\left(\frac{d^2x}{d\theta^2}\right) \quad \ldots\ldots (22)$$

In Equation 22, $dx/d\theta$ and $d\delta/dx$ have been established previously. It is necessary therefore to evaluate $d^2\delta/dx^2$ and $d^2x/d\theta^2$ to complete Equation 22:

$$\frac{d^2\delta}{dx^2} = \frac{d}{dx}\left(\frac{d\delta}{dx}\right) = \frac{d}{dx}\left(\frac{c(\cos\delta) - x}{xc\sin\delta}\right)$$

$$= -\left[\frac{2}{x}\,\frac{d\delta}{dx} + \frac{1}{xc\,\sin\delta} + (\cot\delta) \times \left(\frac{d\delta}{dx}\right)^2\right] \quad \ldots\ldots (23)$$

Also

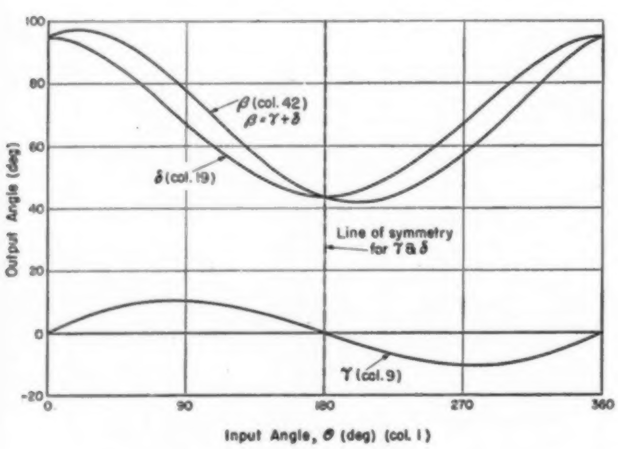


Fig. 5—Symmetrical angular displacement curves for oscillating beam (γ) and crank and connecting rod (δ), and resulting nonsymmetrical curve for four-bar linkage (β)

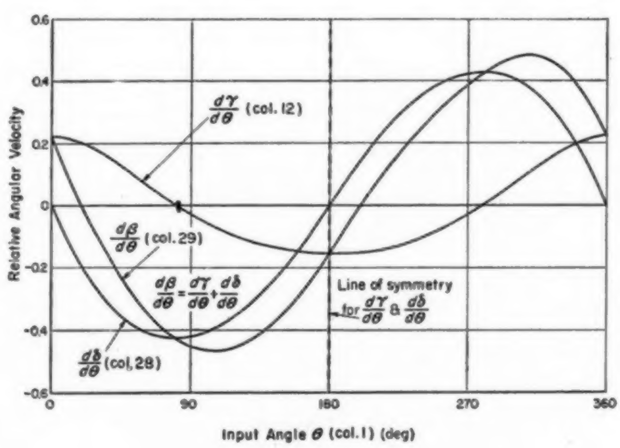


Fig. 6—Angular velocity curves corresponding to displacement curves in *Fig.* 5

$$\frac{d^2x}{d\theta^2} = \frac{d}{d\theta}\left(\frac{dx}{d\theta}\right) = \frac{d}{d\theta}\left(\frac{ad \sin\theta}{x}\right)$$

$$= \frac{ad\cos\theta - \left(\frac{dx}{d\theta}\right)^2}{x} \quad \ldots\ldots\ldots\ldots (24)$$

After substitution of Equations 23 and 24 into Equation 22 the final acceleration coefficient is

$$\frac{d^2\delta}{d\theta^2} = -\left(\frac{dx}{d\theta}\right)^2\left[\frac{3}{x}\frac{d\delta}{dx} + \frac{1}{xc\sin\delta} + (\cot\delta)\left(\frac{d\delta}{dx}\right)^2\right] + \frac{d\delta}{dx}\frac{ad\cos\theta}{x} \quad \ldots (25)$$

Equations 19, 21 and 25 represent the complete relative kinematics of the four-bar linkage. The absolute kinematics of the output can now be established with relative ease from Equations 1 and 2. The absolute velocity of the output as given by Equation 1 is

$$\frac{d\beta}{dt} = \frac{d\beta}{d\theta}\frac{d\theta}{dt}$$

where $d\beta/d\theta$ was determined in Equation 21 and $d\theta/dt$ is absolute angular velocity of the input in radians per sec. The general expression for the acceleration of the output was given in Equation 2 and it applies to accelerated inputs. However, Equation 2*a* is of more practical use in the overwhelming number of dynamic analyses, because it applies to a constant velocity of the input.

Tabulating Scheme: The tabulating scheme applied to the four-bar linkage makes use of the important fact that the oscillating beam and crank and connecting rod are symmetrical mechanisms. Hence, calculating work is practically cut in half. This can be understood by realizing that a four-bar linkage is inherently a nonsymmetrical mechanism and requires an analysis over the entire driver cycle. Using the approach described in this article eliminates the necessity of full cycle analysis except in the final output columns for displacement, velocity and acceleration.

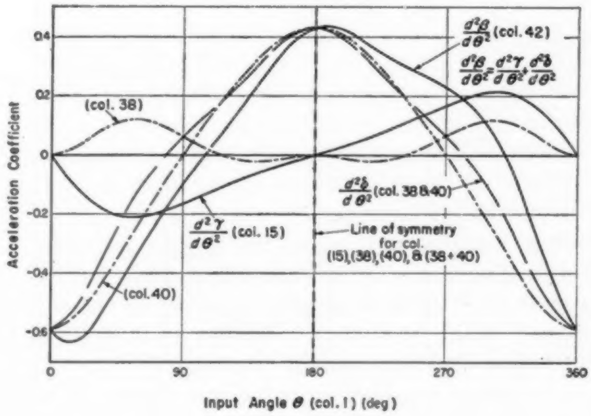


Fig. 7—Angular acceleration curves corresponding to displacement and velocity curves, *Figs.* 5 and 6

KINEMATIC ANALYSIS

The work sheet system shown in *Fig.* 4 is designed for ease in calculating the kinematics described. It is anticipated that some sort of calculating machine will be used where accuracy is required. A slide rule, however, will give results comparable to the graphical solutions. Each column in the table calls for a single operation, many of which are simply multiplication by or addition of a constant. This scheme permits rapid calculations without accumulative errors.

Velocities and accelerations accurate to a fraction of one per cent are attainable by using four decimal places for trigonometric functions and no more than five significant figures for all other functions. Angles should be established to the nearest minute.

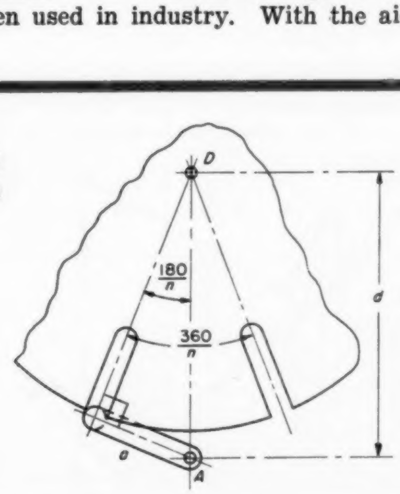
The top section in each column heading is the column number given in parentheses to distinguish it from a numerical value. The middle section is the function itself giving the column relationship to the equations. The bottom section indicates the operation to be performed in the column. Space for the numerical values for the constants involving *a, b, c* and *d*, as defined by *Fig.* 1, is provided where necessary.

The following columns give the end results for the stated mechanisms.

Col.	Equation	Factor
(9)	10	Displacement angle for oscillating beam
(12)	11	Angular velocity ratio for oscillating beam
(15)	12	Acceleration relation for ocsillating beam
(29)	21	Velocity ratio for the four-bar linkage
(41)	25	Acceleration coefficient for the four-bar linkage
(42)	18	Displacement for four-bar linkage

The example worked out in *Fig.* 4 represents the complete kinematics of a four-bar linkage which has actually been used in industry. With the aid

Fig. 8 — External Geneva mechanism



of the tabulating scheme, the actual calculating work has been practically cut in half, due to the fact that nearly all the columns are symmetrical around driver crank angle $\theta = 180$ deg.

Referring to the symbols used in *Fig.* 1, the dimensions of the linkage members are: crank, a = 0.9235 inch; connecting rod, b = 4.756 inches; crank, c = 2.000 inches; center distance, d = 5.077 inches.

The operational steps used in calculating the kinematics of the output crank are easily followed in the table. Values inserted in the column heads for this example are indicated by underline. The 36 points in column (1) were taken for 10 deg increments around the driver cycle. Columns (2) and (3) were filled in for 10 points only, columns (4), (5), (10), (13), (26) were filled in for 19 points, but due to their symmetry around the 90-deg input angle θ only 10 points were calculated and the rest were merely filled in with their proper signs. All other columns with the exception of 2ν 41 and 42 were calculated for only 19 points.

Symmetry of the outputs of the simpler components of the four-bar linkage—the oscillating beam γ and the crank and connecting rod δ around the 180-degree input angle θ is illustrated graphically by the kinematic curves in *Figs.* 5 to 7. *Fig.* 5 shows angle δ to be completely symmetrical while γ is equally so except for a sign reversal. *Figs.* 6 and 7 show the same characteristics. On the other hand, β in *Fig.* 5, $d\beta/d\theta$ in *Fig.* 6 and $d^2\beta/d\theta^2$ in *Fig.* 7, representing the kinematics of a four-bar linkage, are nonsymmetrical.

Tables like *Fig.* 4 can be drawn up for the crank and connecting rod mechanism using Equations 14, 15 and 17.

Geneva Mechanism: *Fig.* 8 represents an external Geneva. The kinematic equations for the Geneva wheel during the active part of the cycle are the same as those derived for the oscillating beam in *Fig.* 2. For a given number of slots, n, in a Geneva wheel, the relation between dimensions a and d is

$$a = d \sin \frac{180}{n} \qquad (26)$$

Now, the kinematic equations for the oscillating beam mechanism apply to the external Geneva for crank angles θ between the limits of 0 and $\pm(90-180/n)$ and the internal Geneva for crank angles θ between the limits of 180 and $\pm(90-180/n)$ deg.

As an example of Geneva application, suppose the kinematics for a 6-slot external Geneva is required. From Equation 26

$$d = \frac{a}{\sin \frac{180}{6}} = \frac{a}{\sin 30} = 2a$$

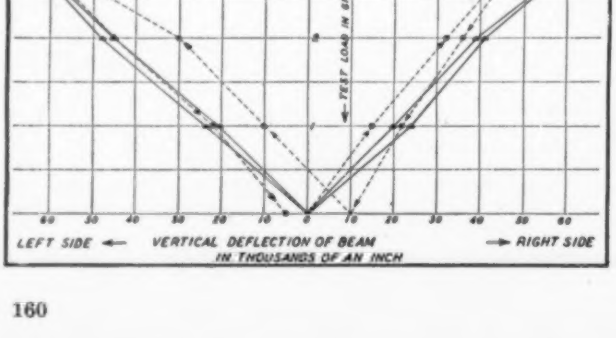
The limits for the crank angle θ become 0 to $\pm(90-180/6)$ deg or 0 to $\pm$ 60 deg.

The analysis made in this article has been restricted to that of the four-bar linkage and its associated simpler mechanisms. The method used, however, can be expanded in an orderly fashion to many combinations of linkages. In this case, it is merely necessary, in making the analysis, to use the output of the first linkage as the input to the next. This process can be continued until any desired motion is determined.

Solid Lubricant Reduces Ball Bearing Friction

LUBRICATION of ball bearings with grease or oil in applications where starting friction and friction at very low speeds must be low, results in higher friction than the dry bearing would produce. One such application is in a Baldwin-Lima-Hamilton spring tester of 5 pounds capacity, in which six precision ball bearings serve as the pivots of an equal arm balance. Recent tests indicate that a solid lubricant will reduce friction values in such service to less than those of the dry bearing.

Fig. 1—Dotted lines show results with unlubricated bearings. Solid lines show results with molybdenum-disulfide lubricated bearings



In the tests, the beam was balanced with 5 pounds on each of the two bale rods. Weight of 3 grams was then added to the right bale rod in increments of 1 gram; weights were then removed in increments and added to the left bale rod, readings being taken of the deviation of the beam from horizontal in 0.001-inch units.

Broken lines, *Fig.* 1, show the results with dry bearings, divergence of the lines indicating a total zero shift of 0.015-inch due to friction of the dry bearings. Solid lines show results obtained using molybdenum-disulfide treated bearings.

Dry bearings were prepared by thoroughly washing in solvents and drying. The other bearings were thoroughly washed in solvent; Molykote Type G molybdenum disulfide lubricant applied and bearings run at 2500 rpm for one minute. This was followed by a thorough solvent washing to give a completely dry bearing with a thin coating of molybdenum disulfide on all bearing surfaces.

Calculating Tearout Strength for Cantilever Beams

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THERE ARE many instances in which cantilever beams carrying high axial bending loads are supported on relatively thin bases. Under these conditions, shear tearout failure of the base is likely to occur. Some examples of where cantilever beam, shear-tearout failure may happen include a beam welded in a shallow socket, axle jack-pad under side load or a lug supported by a thin tube such as employed in the design of a landing gear shock strut, *Fig.* 1.

Generality of the elementary expression My/I when applied to the integration of linear functions is well recognized in connection with the calculation of axial bending stresses. However, it is also the basis for the evaluation of shear stresses with linear variation in this data sheet.

With reference to *Fig.* 2, there is the basic relationship

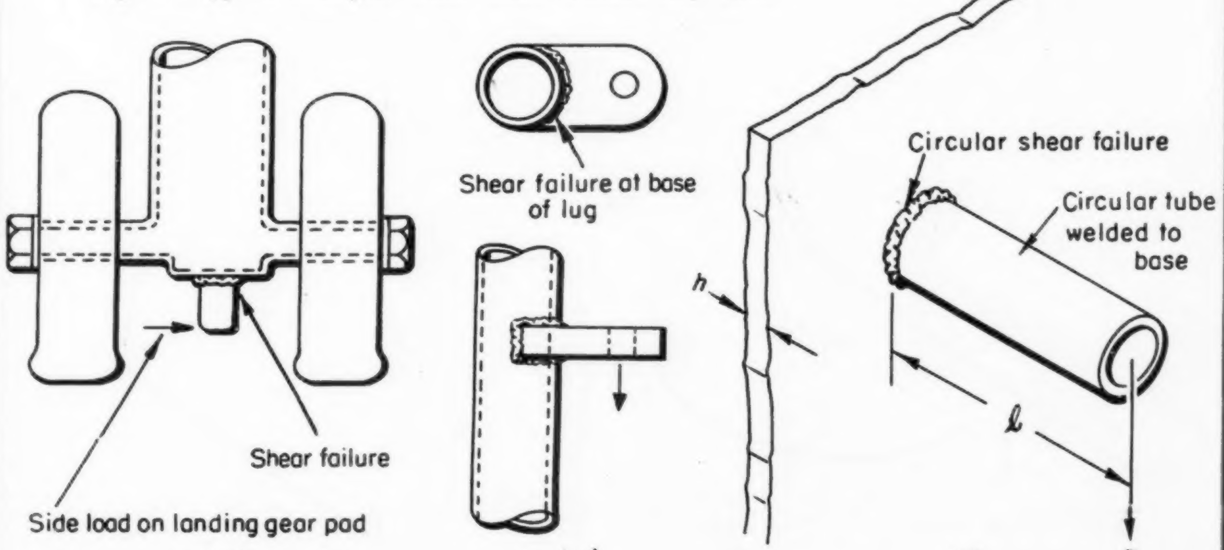
$$Pl = M = \int_c S_t \, y \, dA \qquad (1)$$

where

$$S_t = \frac{r \sin \theta \, S_s}{y_{max}} \qquad (2)$$

for an assumed linear distribution of stress across

Fig. 1—Typical examples where shear tearout may occur



the section *Fig. 2b*. Also from *Fig. 2c*,

$$dA = hds = h\sqrt{r^2 + \left(\frac{dr}{d\theta}\right)^2}\, d\theta \quad \text{(3)}$$

and from *Fig. 2b*,

$$y = r \sin \theta \quad \text{(4)}$$

Substituting the values in Equations 2, 3 and 4 into Equation 1 shows the expression for the shear resisting moment between the joint and the beam to be

$$M = \frac{S_s\, h}{y_{max}} \int_0^{2\pi} r^2 \sin^2 \theta \sqrt{r^2 + \left(\frac{dr}{d\theta}\right)^2}\, d\theta$$
$$= S_s\, h\, C \quad \text{(5)}$$

Nomenclature

A = Area over which shear force acts, in.^{2}
a = Length of semiminor axis of ellipse, in.
b = Length of semimajor axis of ellipse, in.
C = Beam shape factor, in.^{2}
h = Thickness of supporting base of beam, in.
I = Moment of inertia, in.^{4}
l = Beam length, in.
M = Applied moment, lb-in.
P = Load, lb
r = Radius to outside contour of beam, in.
S_s = Allowable shear stress, psi
S_t = Shear tearout stress, psi
s = Arc length, in.
y = Distance from point on beam to neutral x-axis, in.
θ = Angle corresponding to r, radians
σ_b = Bending stress, psi
$\phi(\theta)$ = Contour function for beam shape

where C equals the value of the integral divided by y_{max}. Solving for h gives

$$h = \frac{M}{S_s C} \quad \text{(6)}$$

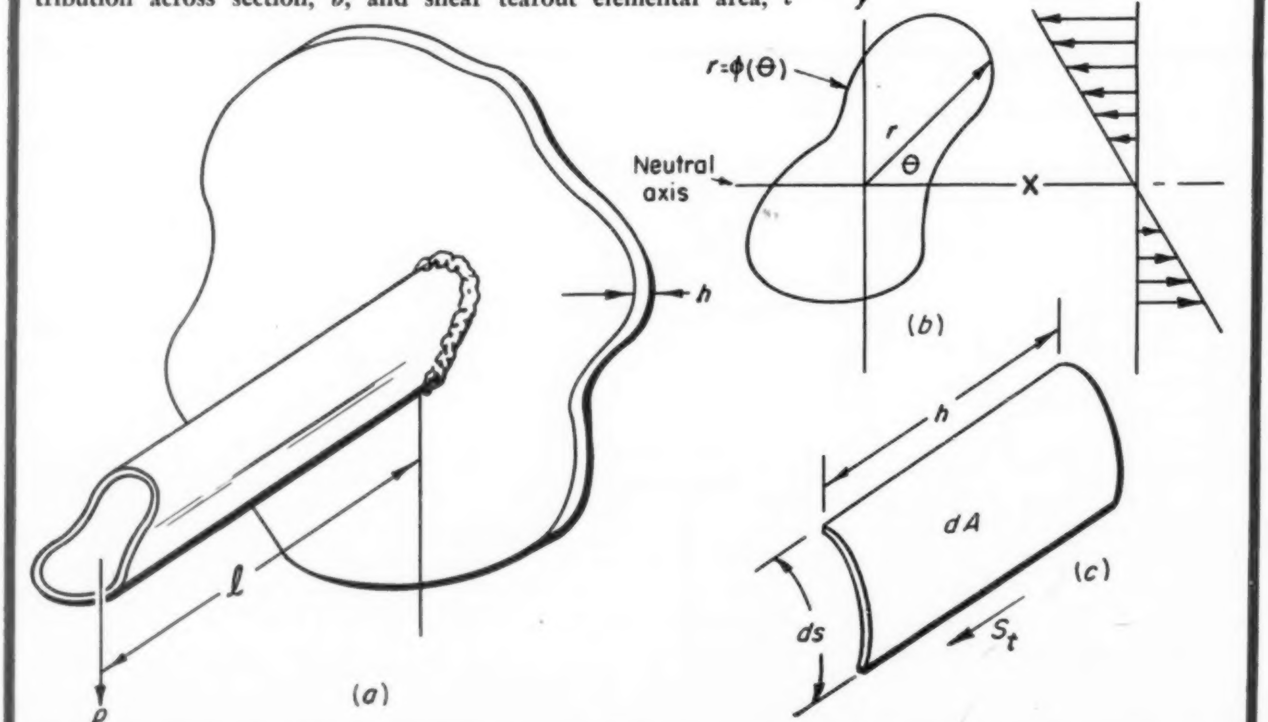
Thus base plate thickness and/or cantilever beam wall thickness required to prevent shear tearout can be determined from Equation 6.

Example 1: Assume that a 347-1A weldable, stainless-steel tube is welded to a socket as illustrated in *Fig. 1c*. The tube has a radius r = 0.5-inch and a bending moment M = 5000 lb-in. For an allowable shear stress S_s = 40,000 psi, from Equation 6, h = 0.16-inch, the thickness required for the base. Similarly for a bending stress σ_b = 75,000 psi, from Equation 6, h = 0.085-inch, the minimum wall thickness of the tubing. For this type of stainless steel tubing of 1 inch diameter, the only standard tube wall-thicknesses are 0.035 and 0.12-inch; hence the 0.12- inch wall-thickness tubing is chosen. For purposes of weight reduction, the tube could be turned down to the calculated 0.085- inch wall thickness.

Equations for computation of C and curves for determining its value directly are given in charts, *Fig.* 3, for circular, rectangular, elliptical, and I or channel-shaped cantilever beam sections. Similar equations and curves could be determined for other beam shapes. Then from Equation 6 base plate thickness or wall-thickness of tubes in cantilever beams required to prevent shear tearout can be quickly evaluated.

Limit Design: Stress is considered constant and maximum at every point across the cantilever beam

Fig. 2—Axial bending load applied to cantilever beam welded to thin metal base, *a*. Beam contour and assumed linear stress distribution across section, *b*, and shear tearout elemental area, *c*



section in limit design. Therefore in Equation 2, $r \sin \theta / y_{max} = S_t = S_s$. It follows then that

$$S_s = \frac{M}{h \int_0^{2\pi} r \sin \theta \sqrt{r^2 + \left(\frac{dr}{d\theta} \right)^2} \, d\theta} \qquad (7)$$

Solving for h for circular shear failure in limit design gives

$$h = \frac{M}{S_s 4r^2} \qquad (8)$$

For cantilever beam of a rectangular section

$$h = \frac{M}{S_s \left(\frac{d^2}{2} + bd \right)} \qquad (9)$$

and for an I-section or channel

$$h = \frac{M}{S_s \left(\frac{d^2}{2} + 2\,bd \right)} \qquad (10)$$

CANTILEVER BEAM TEAROUT

In the case of an ellipse in limit design for loads parallel to major axis b

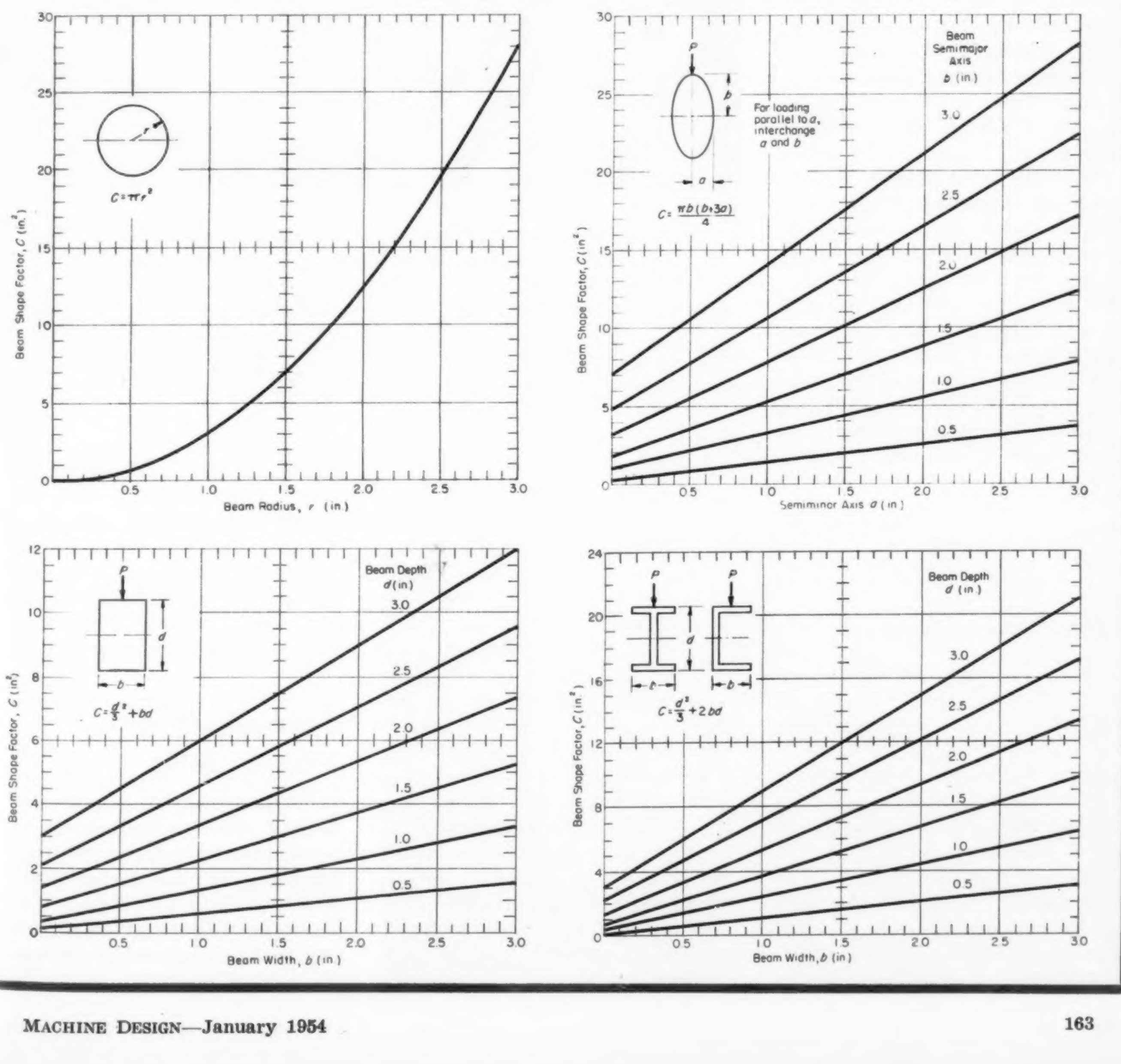
$$S_s = \frac{M}{4\,h \int_0^{\pi/2} b \sin \theta \sqrt{b^2 \cos^2 \theta + a^2 \sin^2 \theta} \, d\theta} \qquad (11)$$

Solving Equation 11 and rearranging gives

$$h = \frac{M}{S_s 2b \left(b - \frac{a^2}{\sqrt{b^2 - a^2}} \log_e \frac{a}{b + \sqrt{b^2 - a^2}} \right)} \qquad (12)$$

Limit design, beam shape factor curves, similar to those in *Fig.* 3, can be easily set up if required and solutions quickly obtained from Equations 8, 9, 10, or 12.

Fig. 3—Equations and curves for evaluation of beam shape factor C for typical beam cross-sections



Engineering NEWS ROUNDUP

Scientists Vindicate NBS Stand on AD-X2

Last spring, Secretary of Commerce Sinclair Weeks asked Dr. Detten W. Bronk, president, National Academy of Sciences to appoint a committee to appraise the quality of work done by the National Bureau of Standards in relation to a battery additive known as AD-X2 which its makers claimed would extend the life of lead-acid storage batteries, among other things. The conclusions of the committee of which Dr. Zay Jeffries, vice president (retired) General Electric Co., Chemical Div., is chairman, were recently published.

Findings of the committee, composed of outstanding scientists from the fields of education, engineering and research, were that the quality of the Bureau of Standards' work in the field of lead-acid storage battery testing is excellent and that relevant data available to the committee are adequate to support the position of the Bureau of Standards that AD-X2 is without merit. No additional tests on the merit of AD-X2 were recommended by the committee.

LIGHT ALLOY CASTINGS used in a 1954 Chrysler sedan reduce its weight by approximately 183 pounds. Even wider use of aluminum and magnesium castings in a convertible reduces weight by 225 pounds. Among the parts shown are pistons, oil filter housing, fuel pump, automatic transmission parts, steering column shroud, power steering housing and torque converter parts

ASME Lecturers Honored

Seven engineers, chosen to represent The American Society of Mechanical Engineers as lecturers because of distinction in their respective fields, were awarded certificates of recognition at the president's luncheon at the annual ASME meeting.

Engineers honored were Earle Buckingham, professor emeritus in the Department of Mechanical Engineering, Massachusetts Institute of Technology, for his lecture, "Dimensions and Tolerances for Mass Production;" Jacob P. Den Hartog, professor of mechanical engineering, Massachusetts Institute of Technology, for his lecture, "Vibration;" Clayton O. Dohrenwend, research engineer, Department of Mechanics, Rensselaer Polytechnic Institute, for "One Hundred Years of Scientific Achievement in Mechanics of Solids;" Charles Lipson, engineering consultant, Detroit, for his lecture, "Stress and Vibrations Problems in Industry;" Glenn Murphy, head of the Department of Aeronautical Engineering, Iowa State College, for his lecture, "Problems of Nuclear Power Development;" Jesse Ormondroyd, professor in the College of Engineering, University of Michigan, for his lecture, "Vibration Measuring Instruments;" and Irwin Vigness, head of the Shock and Vibration Branch, Naval Research Laboratory, Washington, D. C., for his lecture, "Problems of Mechanical Shock and Vibration of Concern to the Armed Forces."

Engineering Makes Television Debut

Engineering has entered the field of adult education via television in the telecourse "Engineering: Building the Modern World," produced by the University of Michigan, College of Engineering. Six leading engineering professors have been teaching the 15 half-

hour lessons designed to present information understandable to the layman.

Carried on TV stations in Detroit, Lansing and Kalamazoo, the course has included lessons on the principles of flight and streamlining, automatic controls, vacuum tubes and transistors, petroleum refining, chemical processing, diesel and internal combustion engines, nuclear engineering, metal castings, strength of metals, and food preservation by irradiation.

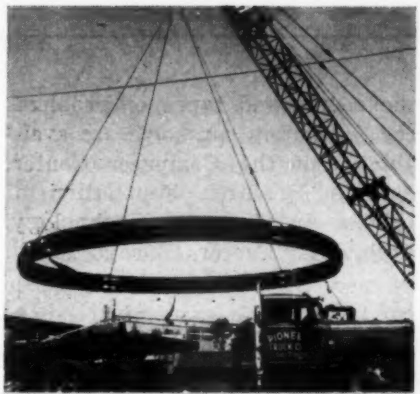
Dr. Robert R. White, professor of chemical and metallurgical engineering, coordinator of the telecourse, says "Engineering is the art and science by which the properties of matter and the sources of power in nature are used in structures, machines, and manufactured products, and we propose to show in the telecourse exactly how this is done. We are going to focus most of our attention on fundamentals and show problems that come up when we have to use fundamentals. Engineering is always a mixture of art and science, plus intuition and guesswork."

Briton Says British Jets Better

Great Britain is five years ahead of the United States in jet engine development according to Dr. Owen A. Saunders, member of the air gas turbine collaboration committee for British aero engine firms, member of his government's Aeronautical Research Council and professor of mechanical engineering at the Imperial College of Science and Technology, University of London. He attributed his country's lead in jet engine development to continuous work in that field since 1943, while U. S. efforts were initiated at a later date. Collaboration among jet engine manufacturers in Britain is also a factor in aiding jet engine development.

Certain other opinions were also expressed by Dr. Saunders in an interview at Illinois Institute of Technology where he appeared as guest lecturer on heat transfer recently. Jet planes will fly at twice the speed of sound in ten years, he predicts, and the need for military pilots will cease by the end of one more generation. Guided missiles will presumably be responsible for this "technological unemployment."

Jets in commercial aviation will be solely for prestige with propeller turbine engines supplanting reciprocating engines in the future. Commercial pilots will still be needed because of the emphasis on reliability and safety in civil aviation, Dr. Saunders said.



BIGGEST BELLOWS ever manufactured are stainless steel expansion joints for a supersonic wind tunnel being built for NACA at Cleveland. A total of 13 expansion joints, ranging from 5 feet to the 28-foot diameter joint shown here were built for the project by Solar Aircraft Co.

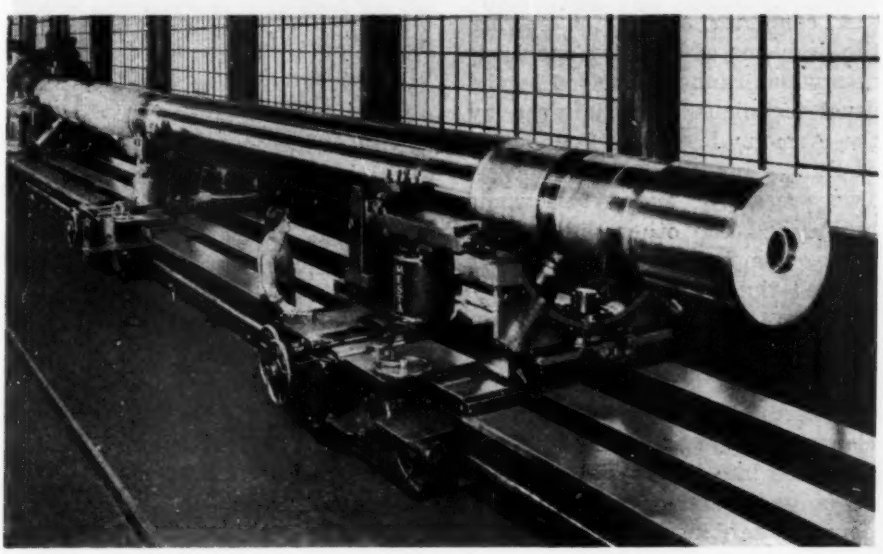
Steel Use Reflects Growth of Western Industry

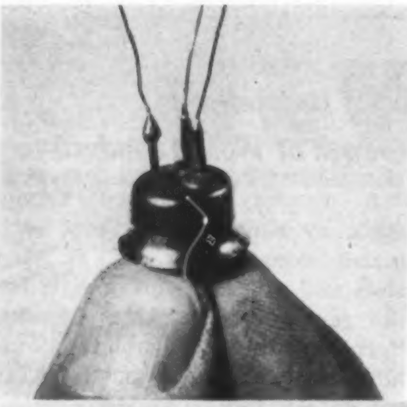
A recently published survey made by Kaiser Steel Corp. estimates consumption of finished steel mill products during 1953 for the seven western states of California, Oregon, Washington, Idaho, Arizona, Nevada and Utah at 6.3 million tons. This exceeds the previous record high of 5.98 million tons for 1951. Total U. S. consumption during 1953 has been estimated at 81.48 million tons which is also an all-time high. Consumption in the seven Western states during 1953 is approximately 7.75 per cent of this total.

During 1952, California, Oregon and Washington used 5.13 million tons of steel mill products. This amount is 7.9 per cent of the U. S. total for '52. In 1947 the same three states used approximately 3.3 million tons or 5.25 per cent of total U. S. production.

A subregional breakdown of consumption in the seven western states for 1952 shows that 80

COLOSSAL COLUMN being turned here on an equally large lathe is one of eight to be used in 30,000-ton Mesta Machine Co. hydraulic die forging press. Forged from a 500,000-pound ingot, the column is 76 feet long and is being turned to diameter of 40 inches on a 96-inch capacity Mesta lathe





SUPER TRANSISTOR is 100 times more powerful than present commercially available models. Low power-handling capabilities have been one of the chief drawbacks of transistors, preventing their use in many applications. Twenty-watt output of this Minneapolis-Honeywell developed transistor is enough to operate motors, valves, relays and other equipment

per cent of the total was used in California, 15 per cent in Oregon and Washington, and the remaining 5 per cent was distributed among Arizona, Nevada, Utah and Idaho. Mills located in these seven Western states provided 58 per cent of the steel used by them in 1952.

A computer center for solving the problems of industry and business which cannot be solved by the usual techniques is maintained by the Armour Research Foundation of Illinois Institute of Technology. Two recently added electronic brains and the other digital and analog computing equipment have enabled the Foundation to reduce waiting time for the start of work on a problem to one month.

Established in 1950, the center is staffed by engineers, mathematicians and physicists who translate problems into the special language of a particular computer. Brochures describing the equipment at the center and types of problems the computers can solve are available from the Computer Center, Armour Research Foundation of Illinois Institute of Technology, Technology Center, Chicago 16, Ill.

Awards Presented At ASME Meeting

Three awards established by The National Machine Tool Builders' Association were presented for the first time at the honors luncheon held at the recent annual meeting of The American Society of Mechanical Engineers.

Machine Tool design and economic value awards were awarded to Ernest Wildhaber, mechanical engineer at the Gleason Works, Rochester, for his paper, "Manufacture and Application of Gleason Toothed Face Couplings and Clutches;" to Robert R. Slaymaker, professor of machine design at Case Institute of Technology, and consultant for Cleveland Graphite Bronze Co., for his paper, "Bearing Design Using Concentric Journal Theory," and to Carroll R. Alden, research engineer, Ex-Cell-O Corp., Detroit, for his paper, "Electrospark Machining."

Gannt Medal of the society was awarded to Thomas E. Millsop, president, Weirton Steel Co., Weirton, W. Va. on behalf of the ASME and the American Management Association. Mr. Millsop was cited as "industrialist, educator, humanist and public official whose life has been dedicated to selfless service to others."

Steel Founders' Announce Product Development Contest

A total of $3500 in prize money will be awarded to winners of a Product Development Contest, announced by the Steel Founders' Society of America, for new applications and improved designs of steel castings. Entries must be submitted by November 1954 and must have been developed since January 1, 1951.

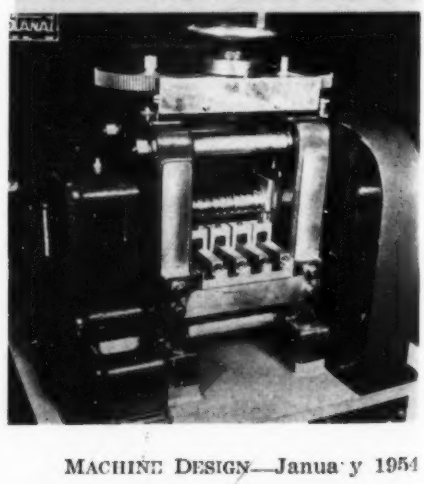
Eligible contestants have been divided into two classes. Class 1 includes all employees of producers of steel castings in the United States, Canada and Mexico except members of the National Product Development Committee of the Society. Class 2 includes all other persons, such as design engineers, employees of customers of the steel castings industry, and college students.

Complete details are obtainable from Steel Founders' Society of America, 920 Midland Bldg., Cleveland 15, Ohio.

More Materials Decontrolled

Revocation of a regulatory measure titled "Designation of Scarce Materials 1" was announced recently by the Business and Defense Services Administration, Department of Commerce. Designation of Scarce Materials 1 was issued by the National Production Authority, predecessor agency to BDSA, to carry out the antihoarding provisions of the Defense Pro-

MINIATURE MILL: Accurate and rapid tube reducing and tube sizing is said to be possible with new small rolling mills having 3 and 4-inch diameter rolls. One of the applications for which the small mill made by Stanat Mfg. Co. is well suited is the processing of tubular heating units in which a refractory powder must be compacted around a resistance element



duction Act. The document specified as scarce the strategic alloying materials, chromium, cobalt, columbium and tantalum, molybdenum, and nickel, and also diamond grinding wheels, all of which are now free from government controls on use, allocation and inventory except for military and atomic-energy orders.

Enter— The Color Engineer

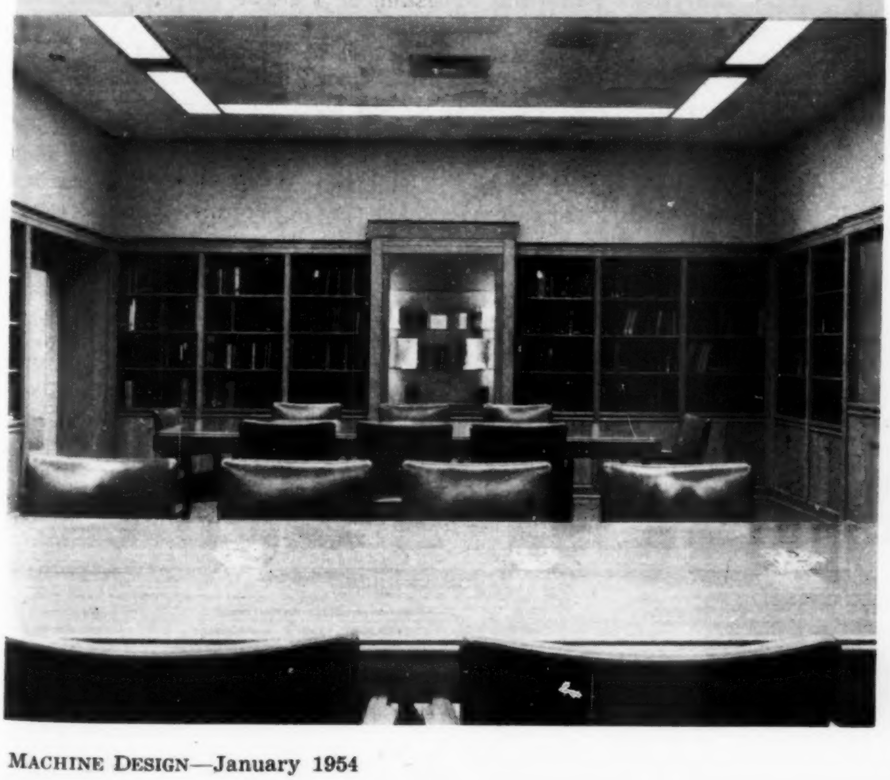
The color "engineer" is putting sleight-of-hand illusions to work to make your car more appealing. V. M. Exner, director of styling, Engineering Div., Chrysler Corp., reports that color has become an important sales factor—making an automobile appear larger, longer, wider and lower.

Warm colors—ivory, cream, red, orange and yellow—will make a car appear larger, added length can be achieved by using horizontal chrome stripes on dark colored cars, and light colors give a wider effect. Two-toning lowers the appearance of the car and the use of lighter colors beneath the belt line and darker ones above make the car appear lower and longer.

Trends for color in automobiles are predicted by studying color trends in other industries, in women's fashions, interior decorating, furniture and foreign markets.

Color preferences vary according to age groups, regions and income. Seasonal preferences also follow a pattern. It has been found that persons who live in the sunny areas (Texas, California and Florida) prefer brighter colored cars while those who live in Northern areas (Seattle, New York) prefer the darker blues, greens and grays. As for income, darker tones are preferred by those in the higher income brackets. Mr. Exner observed that a limousine will probably never appear in a sporty yellow color!

ENGINEERING HISTORY: About 4300 volumes, including 15 books from the 16th century, 33 from the 17th and 154 from the 18th century, make up the William Freeman Myrick Goss Memorial Library of Engineering History at Purdue University. Although the books are not loaned, they are available for use in the library by the general public as well as students and staff



Ordnance Asks For Light, Compact Equipment

A recent bulletin issued by Major General E. L. Ford, Chief of Ordnance points out that the next war, if any, will probably be won or lost quickly. Extreme mobility will be required of our armed forces—the capacity to deliver relatively large forces of men rapidly to any part of the world, despite mud, mountains, morasses or lack of roads. Such mobility is only attainable with an air-liftable soldier, a soldier with light, compact equipment. A soldier's equipment today is not merely a rifle and ammunition. Trucks, jeeps, large-caliber guns and a host of other items make up a soldier's tools.

To aid designers of Ordnance material in producing such equipment, the Ordnance Department has published pamphlets titled "Air Transportability of Materiel" and "Limitations of Materiel for Air Transport." Additionally, a "Guide to Manufacturers in Developing Light and Compact Army Equipment" is currently being circulated to the Staff for comment.

Suggested methods for lightening Ordnance materiel include greater use of aluminum, magnesium, titanium and high-strength steels. More exact stress analyses, new ideas and ingenious uses of both old and new materials should also help to achieve the desired result.

Redesign of representative items is now being contemplated. Department of Defense authorities have assured the Army that both magnesium and aluminum will be made available for justifiable uses, if the requirement is made known sufficiently in advance.

Addition of a swivel fitting department has been announced by Emsco Mfg. Co. Ball bearing swivel fittings will be manufactured by the new department.

Rounding Up the 1954 Automobiles

With four automakers yet to be heard from, the trend to overhead valves, V-eight and higher horsepower seems to be continuing. Two new overhead-valve V-eight engines have been introduced (Mercury, Buick) to bring the total number of such engines used in the cars reported here to nine in contrast to three L-head in-line eights, two overhead-valve in-line sixes, and thirteen L-head in-line sixes. Only three makes of automobiles are not making higher horsepower engines available in their 1954 models. Horsepower increases range from 5 to 55 horsepower in previously manufactured engines and have been obtained by raising compression ratios, improved manifolding, improved valve gear, dual exhausts and more-barrel carburetors.

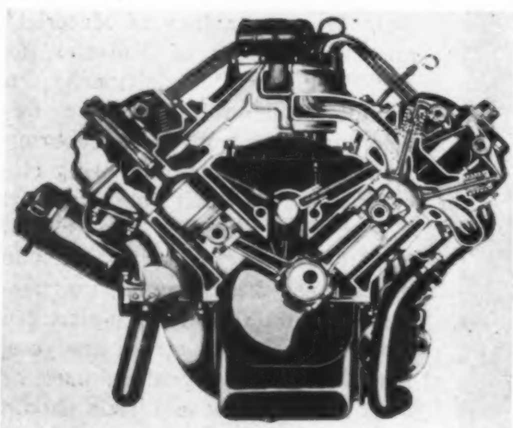
Styling or body changes are primarily of the face-lifting variety. Buick's wraparound windshield and Mercury's transparent top on a sports model are probably the most unusual styling features to appear to date. Much emphasis is being placed upon new, colorful interior fabrics and interior treatments. Nylon, vinyls and vinyl-faced upholstering materials are being used on many of the new cars.

Models not covered here will appear in February.

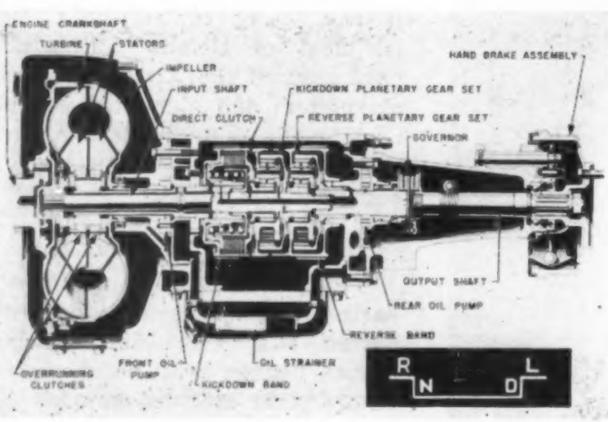
CHRYSLER

OUTSTANDING changes in the '54 Chryslers are more powerful versions of the Firepower V-8, a fully automatic torque converter transmission, new front suspension which reduces body lean on turns, and a smoother, more solid ride resulting from stiffer frame, and new body and rear-spring mounting. Body changes include wraparound rear window plus grill and trim modifications.

Horsepower of the Chrysler Firepower V-8 has been increased from 180 to 195 in the New Yorker and to 235 in New Yorker DeLuxe, Custom and Crown Imperials. Intake valve and port diameters were increased ⅛-inch, and exhaust valve and port diameters by ¼-inch to give the New Yorker its 15 more horsepower. The same valve and port diameter increase combined with four-barrel carburetor, larger air cleaner, larger cross-sectional area of intake and exhaust manifolds, and that favorite of the hot rodders —dual exhausts, add up to 55 more horses for the New Yorker DeLuxe, Custom and Crown Imperials. Dual exhausts result in more efficient silencing, unlike those usually used on the hot-rods.



Chrysler's Powerflite consisting of a torque converter and two-speed planetary transmission, has an overall torque multiplication of 4.47 to 1, claimed to be the highest starting ratio in the industry. Torque converter multiplication is 2.6 to 1 and starting gear ratio is 1.72 to 1. As shown by the inset, the transmission has no parking position and the selector lever is "gated" to make shifting by feel possible. An internal-expanding hand brake at the rear of the transmission is said to eliminate the need for the parking brake. Shifting from reverse to low does not require passing through drive or neutral position; the selector lever when pulled toward the steering wheel can be moved from reverse directly to low or vice versa. This should facilitate "rocking," when necessary. Also available on DeSotos and Dodges, Powerflite is said to weigh less and have fewer parts than other automatic transmissions.



Engine Specifications

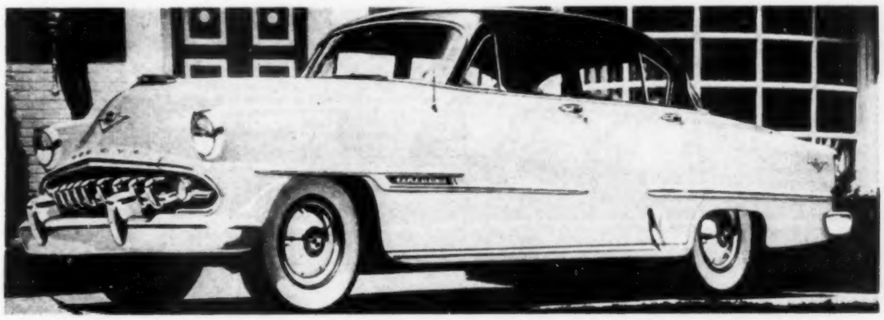
	Windsor	New Yorker	New Yorker Deluxe*
Type	L-head, in-line	OHV, Vee	OHV, Vee
No. cyls.	6	8	8
Bore & stroke (in.)	$3\frac{7}{16}$ x $4\frac{5}{8}$	$3\frac{13}{16}$ x $3\frac{5}{8}$	$3\frac{13}{16}$ x $3\frac{5}{8}$
Displ. (in.^{3})	265	331.1	331.1
Comp. ratio	7.0 to 1	7.5 to 1	7.5 to 1
Bhp, max	119 @ 3600 rpm	195 @ 4400 rpm	235 @ 4400 rpm
Torque, max (lb-ft)	218 @ 1600 rpm	320 @ 2000 rpm	330 @ 2600 rpm

* Also Custom and Crown Imperial

Size

	Windsor, New Yorker	Custom Imperial	Crown Imperial
Wheelbase (in.)	125½	133½	145½
Length (in.)	215⅝	223¾	236[illegible]
Width (in.)	77½	77⅝	[illegible]
Height (in.)	62¾	63	[illegible]

DESOTO



TEN more horsepower, the Powerflite transmission, new front suspension, stiffer frame, redesigned body mountings and new rear spring mountings are incorporated in the new DeSoto. Compression ratio increase from 7.1 to 1 to 7.5 to 1 appears to be responsible for the horsepower increase. New upholstery fabrics, for the most part nylon-faced for durability, are offered in a variety of colors. Power steering and power brakes are available as special equipment.

Engine Specifications

	Fire Dome	Powermaster
Type	OHV, Vee	L-head, in-line
No. cyls.	8	6
Bore & stroke (in.)	3⅝ x 3 11/32	3 7/16 x 4½
Displ. (in.³)	276.1	250.5
Comp. ratio	7.5 to 1	7.00 to 1
Bhp, max	170 @ 4400 rpm	116 @ 3600 rpm
Torque, max (lb-ft)	255 @ 2000 rpm	208 @ 1600 rpm

Size

Wheelbase (in.)	125½
Length (in.)	214½
Width (in.)	77⅝
Height (in.)	62½

DODGE

FASTEST American stock car ever timed by American Automobile Association Contest Board officials is the 1954 Dodge V-8. In October at Bonneville Salt Flats, Utah, a four-door sedan and convertible broke 196 AAA records for distances of one kilometer to 7000 miles. Among other records, the four-door sedan equipped with a conventional transmission established a top speed mark of 108.36 mph on the straightaway. During a 48-hour endurance run the same car achieved lap speeds of 107 mph and averaged better than 101 mph for the entire run including pit stops. The convertible, equipped with the new Powerflite transmission available on 1954 Dodges, ran at full throttle over a 72-hour period and broke more than 60 AAA records formerly held by cars equipped with conventional transmissions.

For '54 the Red Ram V-8 engine has been stepped up from 140 to 150 hp. At least partly responsible is an increase in compression ratio from 7.1 to 7.5 to 1. Powerflite transmission and power steering are available as special equipment. A new super deluxe series known as the Royal V-8, shown, is available as well as Coronet and Meadowbrook V-8 and Six. Horsepower of the 6-cylinder engines has been increased to 110. Much emphasis is placed upon new interior styling and fabrics to harmonize with 11 new body colors and 14 two-tone combinations.

Engine Specifications

	Royal, Coronet 8	Meadowbrook 8	Coronet, Meadowbrook 6
Type	OHV, Vee	OHV, Vee	L-head, in-line
No. cyls.	8	8	6
Bore & stroke (in.)	3 7/16 x 3¼	3 7/16 x 3¼	3¼ x 4⅝
Displ. (in.³)	241.3	241.3	230.2
Comp. ratio	7.5 to 1	7.1 to 1	7.25 to 1
Bhp, max	150 @ 4400 rpm	140 @ 4400 rpm	110 @ 3600 rpm
Torque, max (lb-ft)	222 @ 2400 rpm	220 @ 2000 rpm	190 @ 1600 rpm

Size and Weight

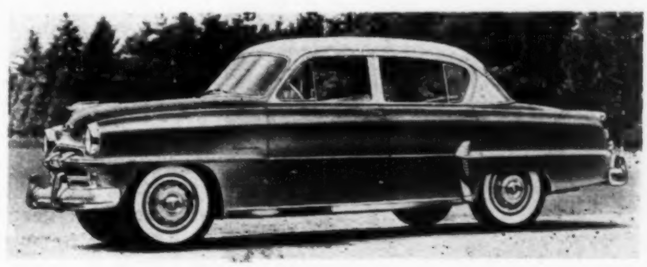
Wheelbase (in.)	119
Length (in.)	205½
Width (in.)	73½
Height (in.)	62
Weight (lb)	3425

PLYMOUTH

POWER steering available on the '54 Plymouths is said to permit turning the wheels on dry pavement when parking with only 6.7 pounds of effort as opposed to the 30.5 pounds required in a similar situation with manual steering. Redesigned shock absorbers and new seat cushion springing are said to give a smoother, softer ride while increased use of silencing materials give a quieter ride. The Hy-Drive introduced in 1953, which is made up of a torque converter and conventional three-speed transmission, is available as well as the overdrive and conventional three speed transmissions. More colorful interiors and more durable, easy-to-care-for fabrics are used in the new models. All models are longer than in '53. A new series, the Belvedere, said to be the aristocrat of the line, has been introduced; other series are the Savoy and Plaza.

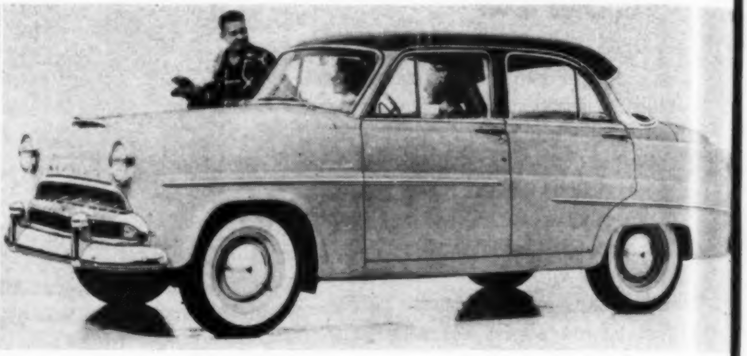
Engine Specifications

Type	L-head, in-line
No. cyls.	6
Bore & stroke (in.)	3¼ x 4⅝
Displ. (in.³)	217.8
Comp. ratio	7.1 to 1
Bhp, max	100 @ 3600 rpm
Torque, max (lb-ft)	177 @ 1200 rpm



Size

Wheelbase (in.)	114
Length (in.)	193½
Width (in.)	74¼
Height (in.)	64



HUDSON

BODY and trim changes, increased power and performance, power steering, an improved power braking system, and salon-lounge interiors harmonizing with exterior colors are incorporated in the new Hudsons. Horsepower and torque increases in Wasp, Super Wasp and Hornet engines are the result of compression ratio increases, a new long-dwell camshaft and combustion chamber redesign.

Power steering, optional at extra cost, is the linkage type. Spring resistance of 4 pounds provides road feel, although as much as 80 per cent of required steering effort is provided by the power steering system. Steering ratio with power steering is the same as with the conventional gear.

Two safety measures are incorporated in the power braking system. A reserve vacuum tank permits up to three power brake applications. Direct mechanical linkage will operate the brakes hydraulically without power assistance.

Redesign of trunk lids, rear fenders, grills and bumpers and the addition of a rub rail change the appearance of the '54 Hudsons appreciably, although basic body shells remain the same.

Highest power to weight ratio in the light car field is claimed for the 1954 Hudson Jet, Super Jet and Jetliner. Engines have redesigned combustion chambers for quicker acceleration throughout the driving range. An optional aluminum, high compression cylinder head and twin carburetor fuel induction system may be had to increase horsepower from 104 to 114. A new series, the Jet-liner has an extra-lush interior. The two-door Jet Utility Sedan has a removable back seat to provide a large hauling space. Hydra-matic drive is available at extra cost.

Engine Specifications

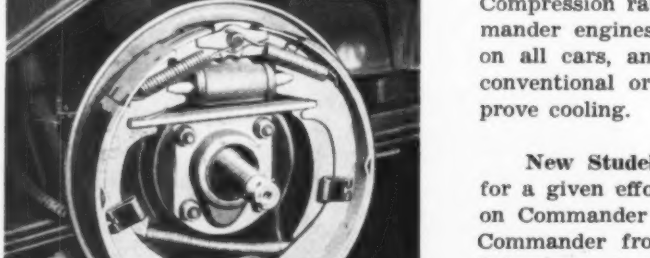
	Jet, Super Jet	Wasp	Super Wasp	Hornet
Type	L-head, in-line	L-head, in-line	L-head, in-line	L-head, in-line
No. cyls.	6	6	6	6
Bore & stroke (in.)	3 x 4¾	3 9/16 x 3⅞	3 9/16 x 4⅜	3 13/16 x 4½
Displ. (in.³)	202	232	262	308
Comp. ratio	7.5 to 1	7 to 1	7 to 1	7.5 to 1
Bhp. max	104 @ 3800	126 @ 4200	140 @ 4000	160 @ 3800
Torque. max (lb-ft)	158 @ 1400			

Size and Weight

	Jet, Super Jet, Jetliner	Wasp	Super Wasp	Hornet
Wheelbase (in.)	105	119	119	124
Length (in.)	180 11/16	201½	201½	208 15/32
Width (in.)	67 1/16	77¼	77¼	77¼
Height (in.)	60⅞	60⅜	60⅜	60⅜
Weight (lb)	2675*	3440	3525	3620

*2725 lb for Super Jet; 2760 for Jetliner.

STUDEBAKER



NAMED the Conestoga in honor of the covered wagons produced 100 years ago or more by the Studebaker family, a new station wagon is the first such model produced by this manufacturer. Like other Studebakers, the Conestoga is styled by Raymond Loewy and follows other models closely in its styling.

The new body type and new brakes are the outstanding features of the '54 Studebakers. Last year's continental styling has been retained with such modifications as new grilles, bumper guards, rub rails and emblems. Compression ratios have been raised to 7.5 to 1 on both Champion and Commander engines. Changed shock absorber valving improves riding qualities on all cars, and new clutch linkages reduce pedal pressure on cars using conventional or overdrive transmissions. New radiator cores and fans improve cooling.

New Studebaker brakes provide up to one-third greater braking power for a given effort by the driver. Front drums on Champion and rear drums on Commander have been increased from 9 to 10 inch diameter. Width of Commander front drums has been increased by ¼-inch. Champion brake lining area has been increased by 3.4 square inches and Commander lining area has been increased 13.4 square inches. The brakes are self-centering and self-energizing, have heavier drums and require only one adjust-

ment per wheel during periodic or relining adjustments. Self-energization of the brakes is derived from the tendency of rotating drum to roll the shoes toward the drum due to the shoe mounting method employed, which uses no rigid anchors.

	Engine Specifications: Champion		Commander, Land Cruiser
Type	L-head, in-line		OHV, Vee
No. cyls.	6		8
Bore&stroke(in.)	3 x 4		3.375 x 3.25
Displ. (in.^{3})	169.6		232.6
Comp. ratio	7.5 to 1		7.5 to 1
Bhp, max	85 @ 4000 rpm		120 @ 4000 rpm
Torque, max (lb ft)	138 @ 2400 rpm		190 @ 2000 rpm
Size	Sedans	Coupes	Station Wagon
Wheelbase (in.)	116.5	120.5	116.5
Length (in.)	198.63	202.22	195.63
Width (in.)	69.5	71	69.75
Height (in.)	59.75	56.31	62.38

NASH


BIGGEST news from Nash is the four-door Rambler, shown, the first four-door available in the Rambler line. Longer wheel base than other Ramblers provides ample rear entrance room, rear leg room and luggage capacity in the new Nash Rambler four-door. Hydra-matic transmission, overdrive, reclining seats and convertible twin beds are offered as optional equipment. Other members of the Rambler line are station wagon, convertible and hard-top. Continental tire mount is standard on Custom four-door convertible and hard-top.

Farina styling has been retained in all series with a few styling refinements. Horsepower has been increased from 100 to 110 for Statesman models by raising compression ratio to 8.5 to 1 and using dual carburetion. Ambassador horsepower is upped from 120 to 130 with an accompanying compression ratio increase. Optional cylinder head and duel carburetion are available to increase Ambassador horsepower to 140. Power steering, power brakes and electric window lifts are offered.

Nash Statesman and Ambassador models have a new concave grille. Continental rear tire mount is standard equipment on all custom four-door and Country Club hard top models. Optional features are Hydra-matic transmission, overdrive, twin beds and reclining seats.

Engine Specifications

	Rambler, 2-Door	Rambler, 4-Door, 2-Door with Hydramatic	Statesman	Ambassador
Type	L-head, in-line	L-head, in-line	L-head, in-line	OHV, in-line
No. cyls.	6	6	6	6
Bore & stroke (in.)	$3\frac{1}{8}$ x 4	$3\frac{1}{8}$ x $4\frac{1}{4}$	$3\frac{1}{8}$ x $4\frac{1}{4}$	$3\frac{1}{2}$ x $4\frac{3}{8}$
Displ. (in.^{3})	184	195.6	195.6	252.6
Comp. ratio	7.25 to 1	7.3 to 1	8.5 to 1	7.6 to 1
Bhp, max	85 @ 3800 rpm	90 @ 3800 rpm	110 @ 4000 rpm	130 @ 3700 rpm
Torque, max (lb-ft)	150 @ 1600 rpm	150 @ 1600 rpm	155 @ 1000 rpm	220 @ 1600 rpm

Size and Weight

	Rambler, 2-Door	Rambler, 4-Door	Statesman	Ambassador
Wheelbase (in.)	100	108	$114\frac{1}{4}$	$121\frac{1}{4}$
Length (in.)	$185\frac{3}{8}$*	$193\frac{3}{8}$	$202\frac{1}{4}$‡	$209\frac{1}{4}$‡
Width (in.)	$73\frac{1}{2}$	$73\frac{1}{2}$	78	78
Height (in.)	59	59	$61\frac{3}{4}$	$62\frac{1}{4}$
Weight (lb)	2550†	2650	3070	3480

* Station wagon, $178\frac{1}{4}$. †Hardtop. ‡Add 10 in. for continental tire mount.

LINCOLN

LINCOLN is standing pat with its 205 horsepower overhead-valve, V-8 engine; no power increase for the '54 models yet. Refinements in carburetor and distributor design, redesigned hydraulic tappets and new fuel filtering elements are said to provide smoother, quieter and more dependable operation. Styling changes include a new hood ornament, repositioned parking and turn indicator lights, restyled side molding and a new rear-quarter gravel shield. Brake lining area has been increased 10 per cent to 220 square inches. Newly designed tires minimize tire squeal on turns. Automatic transmission is standard. Power steering, brakes, window lifts and power adjusted seat are optional equipment.

Engine Specifications

Type	OHV, Vee
No. cyls.	8
Bore & stroke (in.)	3.80 x 3.50
Displ. (in.^{3})	317
Comp. ratio	8.0 to 1
Bhp. max	205 @ 4200 rpm
Torque, max (lb-ft)	305 @ 2300–3000 rpm

Size and Weight

Wheelbase (in.)	123
Length (in.)	214.8
Width (in.)	77.1
Height (in.)	62.7
Weight (lb)	4277

MERCURY

A NEW overhead-valve, V-8 engine developing 161 horsepower shares the distinction of being the outstanding news in the '54 Mercury line with the "Sun Valley," a sports coupe with transparent green-tinted Plexiglass forming a portion of the top. The ball-joint front suspension first used on 1952 Lincolns is also new for '54.

Features of the engine, which develops 36 more horsepower than that used in 1953, are oversquare design (bore larger than stroke) and a stronger, more rigid, five-main-bearing crankshaft with eight counterweights for balancing and smooth operation. A four-barrel carburetor is used.

Conventional kingpins, bushings and upper and lower control-arm outer pivot pins are replaced by two ball joints at each wheel on the new front suspension. Riding and steering qualities are said to be improved by the ball-joint suspension.

Restyled rear quarter panels, new wraparound bumpers, new grille and a new

ornament on the front fenders are all part of the face-lift for the '54 Mercury. Eight new exterior colors will be available as well as new upholstery fabrics.

Power brakes, power steering and power-operated seat are available as optional equipment. Choice of standard, over-drive and Merc-O-Matic transmissions may be made.

Engine Specifications		Size and Weight	
Type	OHV, Vee	Wheelbase (in.)	118
No. cyls.	8	Length (in.)	206.2
Bore & stroke (in.)	3.62 x 3.10	Width (in.)	74.4
Displ. (in.³)	256	Height (in.)	62.2
Comp. ratio	7.5 to 1	Weight (lb)	3620*
Bhp, max	161 @ 4400 rpm		
Torque, max (lb-ft)	238 @ 2200-2800 rpm		

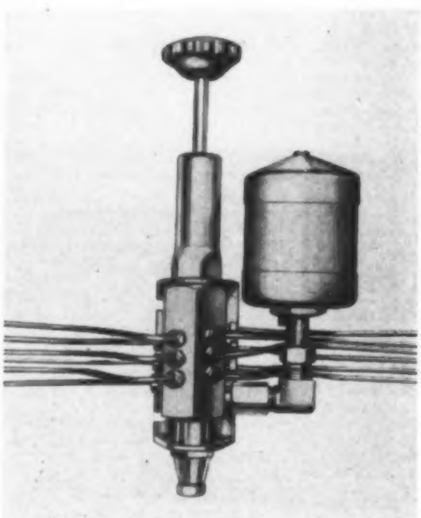
* With standard transmission

To be concluded in February

Centralized Lubrication For Farm Machinery

A newly developed device provides a method of lubricating 12 bearings on a piece of farm equipment, such as a tractor, simultaneously from one fixed location. Supplied in kit form and easily installed, the Multi-Luber eliminates the necessity of periodically suspending operations to grease the bearings.

A piston type manually-operated lubricant pump, lubricant reservoir and tube or hose lines to the bearings make up the complete installation. Construction of the



pump assures that each bearing receives the correct amount of lubricant when the pump is operated despite varying fits of bearings. In the event of lubricant line leaks or breakage, the remaining bearings will still receive the same amount of lubricant.

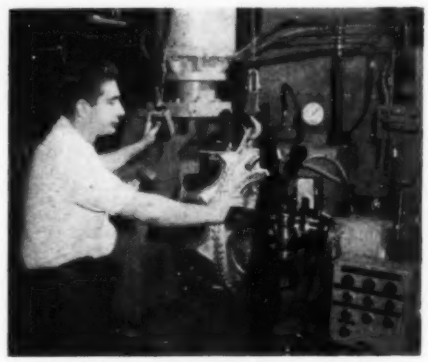
Extreme-pressure gear lubricants are recommended for use in the system rather than greases.

Superior flushing action and best possible lubrication result from the use of the gear lubricants according to the manufacturer, Lincoln Engineering Co.

Automation Applied To Aircraft Engine Assembly

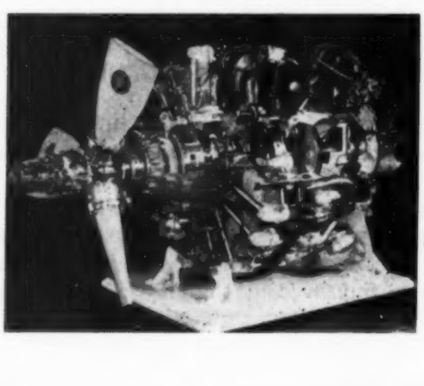
An assembly line incorporating a complete conveyor and belt system plus automatic machines that do everything from assembling a complete crankshaft to tightening bolts and crimping fasteners has the capacity to produce 2½ times as many Curtiss-Wright Turbo Compound engines as the previous assembly line. Space used by the new line is 42 per cent less than used by the line it replaces. Though generous use of automation is made at many points, engines ranging in output from 3250 to more than 3500 horsepower can be built with the same equipment.

Engines are first completely as-

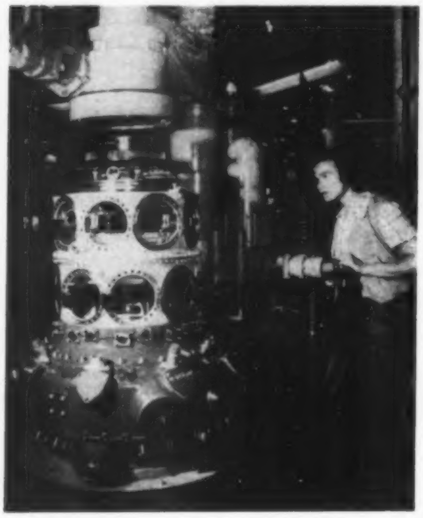


Assembly starts with the crankshaft. Three sections are clamped in this machine and the master and connecting rods are then placed in position. Parts are automatically assembled with proper clearances and bolt torques

Cutaway of Curtiss-Wright Turbo Compound engine. Three exhaust-driven turbine wheels transmit power obtained from exhaust gases to the engine's main shaft through gears and fluid couplings



Conveyor line moves pistons and cylinders to the engine for installation. Engines on vertical stands are clamped to special machines which index engine assemblies and extend the connecting rods for installation of pistons and cylinders



Automatic torque machine tightens and locks the reduction drive gear of the engine's power section

sembled and run for 4½ hours. The engine is then torn down for a complete inspection of all parts. Reassembly of the engine and another 3½ hour test run follow the inspection. If approved, the twice-tested engine is then packed in a sealed container filled with dehydrated air at 7 psi.

Fast-working equipment that is superior to the most highly skilled workmen maintain the high standards of quality during assembly. One machine, for example, runs down the bolts holding a series of pinion gears to identical torque or tightness and then, simultaneously, crimps the locking devices which prevent possible loosening of the bolts.

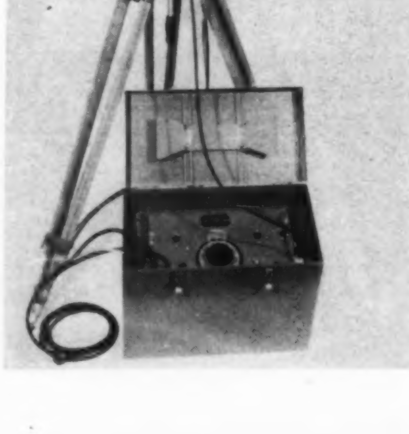
Camera Shutter Speeds Measured in Microseconds

A recently developed camera shutter makes exposure times of one micro-second possible. Known as the Rapatronic shutter, it utilizes the Faraday effect of magneto-optical rotation and has no moving parts. Observation of even the fastest reaction is made possible by the shutter developed by Edgerton, Germeshausen & Greer Inc.

As far back as 1822, Faraday experimented with a beam of polarized light passing through a transparent solution carrying a current to see whether or not this would have any effect on the polarized light. His later work established the existence of the Faraday effect, magnetic rotation of the plane of polarized light passing through an optical element. The shutter using this principle requires crossed polarizing elements and a light transmitting element free from strain. If theoretically perfect polarizers were used, the shutter would reject all light when the magnetic field to rotate the plane of polarization was not applied. Actual polarizers cannot produce perfect light rejection but in the Repatronic camera the closed position is attained. When opened, one-thirtieth transmission is obtained with a three polarizer type lens. This open-to-closed transmission ratio of approximately 30 million is necessary in photographing high light-intensity subjects which persist at high levels of luminosity for long times.

The electromagnetic field strength required to rotate the plane of polarization of the light is quite high. A condenser discharge is used to supply this energy. Light from the subject or a separate flash unit falling on a photocell triggers the condenser circuit.

Rapatronic Camera, Type 2208-0, consists of a 4 x 5-in. Eastman Kodak Master view camera equipped with a Wollenak F/6.3 Raptar lens, the electronic shutter assembly in a 12 x 12 x 3-inch aluminum casting, and a power supply unit in a 16 x 19 x 10-inch aluminum box. Closed-open-closed shutter time is about 0.8 microsecond



Frozen by the Rapatronic shutter in a few millionths of a second, the fiery ball from a nuclear device disintegrates a steel tower shortly after detonation at the Atomic Energy Commission's Nevada proving ground in the spring of 1953. The fireball has already consumed the upper portion of the tower and is seen progressing down the structure

New Fits-Limits Standard

Early this year at the ABC Conference a Draft Proposal for an American, British, Canadian System of limits and fits was accepted by the United States delegation as the basis for a new American Standard if and when developed. The U.S. delegation was made up of members of ASA Sectional Committee B4 and representatives of the U. S. Armed Forces. Present American Standards do not contain tables of recommended fits but state that a study is being made of further recommendations on this subject.

Comments of interested parties on the ABC System are invited by the American Standards Association. Copies of the Draft Proposal are available from the American Standards Association Inc., 70 E. 45th St., New York 17, N. Y. at 75 cents per copy.

Helpful Literature

(*Continued from Page* 178)

35. Bimetal Thermostats

Stevens Mfg. Co.—2-page illustrated data sheet on type C bimetal strip thermostats describes both standard and hermetically sealed types. Operating principle, ratings, performance curves and other technical data are given.

36. Power Cable Handling

Industrial Electrical Works—Various types of Powereel power cable reels are illustrated and described in available catalogs and price lists. Units are available in a range of sizes and mounting styles to suit almost any purpose. Complete specifications are provided.

37. General Purpose Pump

Hypro Engineering, Inc.—Form 2531 illustrates and outlines features of newly developed general purpose pump for industrial and other uses. Unit has 12-vane impeller. Performance tables, drive assemblies and size ranges are described along with various applications.

38. Protected Mercury Switches

Minneapolis-Honeywell Regulator Co., Micro Div.—Protected mercury switches, with glass tube imbedded in solid wax or plastic surrounded by metal or phenolic cases, in a variety of shapes and sizes are shown on data sheet 84-99. Units are available in both ac and dc styles for various applications.

39. Resin Protective Coatings

Bakelite Co.—"Bakelite Resin Coatings for General Industry" is title of 8-page illustrated folder No. VG showing cost savings of various coatings in uses ranging from grain elevators, oil pipes, railroad hopper cars and mining timbers to dairy processing equipment. Coatings resist both corrosion and wear.

40. Stainless Steel Products

G. O. Carlson, Inc.—Stainless steel plates, forgings, No. 1 finish sheets, tank heads and flanges and pattern-cut rings and disks are among the products shown in 4-page illustrated bulletin. Company specializes in fabricating stainless parts in any size and shape.

41. Water Filters

Marvel Engineering Co.—Adaptation of all models of Synclinal filter of sump and line type to water applications is covered in 12-page illustrated catalog 300. Eight sizes from 5 to 100 gpm capacity are detailed. Meshes range from coarse 30 to very fine 200.

42. Time Switches

General Electric Co.—Selection and application information, plus descriptive data, are found in 24-page bulletin GEA-5965 on G-E time switches, process timers and time meters. Operating data and prices are included.

43. Quality in Socket Screws

Parker-Kalon Corp.—Various steps this company takes in maintaining quality control and inspection in manufacture of P-K socket screws are shown in this behind-the-scenes folder, form 481C.

44. Carbides for Wearproofing

General Electric Co., Carboloy Dept.—Use of cemented carbides in wearproofing pulverizing equipment is told in 12-page illustrated bulletin WR-107. Table of standard and special tip shapes for pulverizing blades and hammers is included.

45. Roller Chains & Sprockets

Chain Belt Co., Baldwin-Duckworth Div.—Inherent advantages of roller chain and illustrations of all popular sizes of Baldwin-Rex roller chains are found in 54-page bulletin 52-1. Section is devoted to selection of roller chain drives and has formulas, tables and examples. Characteristics, prices, maintenance and selection of sprockets are covered. Specifications of all units are included.

46. Geared Flexible Couplings

Link-Belt Co. — Line of geared flexible couplings for high speed and high torque applications are covered in detail in 4-page illustrated folder 2375. Ten standard sizes are available with 1 to 6-in. bore sizes and horsepower ratings from 2 to 450 at 100 rpm. Dimensional tables and selection data are included.

47. Pressure Switch

Bobrick Mfg. Corp.—Series 600 multi-range pressure switch, described and illustrated in 4-page bulletin, features high sensitivity, low temperature reliability, durable calibration and low on-off differential pressure at all times. It is designed for controls in medium and high pressure systems using hydraulic fluids, water, air or gas.

48. The Story of Onan

D. W. Onan & Sons Inc.—Over 120 photographs in this 36-page plastic-bound brochure "Measurement Factors of a Company and Its Product" tells the story of Onan products and how they are made. Various models of Onan electric generating plants are illustrated.

49. Brass Sheet & Strip

American Brass Co.—Strong, hard and springy, Formbrite drawing brass retains its ductility in forming and drawing and takes sharp, clear-cut die impressions. It is described in illustrated 12-page booklet B-39. Properties and pressroom and finishing procedures are covered.

50. Gears & Materials Handling

Michigan Tool Co., Cone-Drive Gears Div.—8 illustrated pages of bulletin MH-53 "Cone-Drive Gears at Work in Materials Handling" present 14 typical handling equipment applications of double enveloping worm gear sets and speed reducers. Hoists, cranes, winches and conveyors are covered.

51. Guide To Shell Molding

Durez Plastics & Chemicals, Inc.—Comprehensive 36-page booklet "Durez Guide to Shell Molding" discusses in detail the method, materials and equipment used in this process. It also deals with many common problems and suggests possible causes and solutions.

52. Ceramic Insulators

M. Kirchberger & Co.—4-page "Property Chart" tabulates physical and electrical characteristics of steatite, ceramic and lava insulators for electrical, electronic and chemical applications. Parts can be custom-molded, extruded or machined.

53. Hydraulic Control Panel

Almo Tool Co. — Details of the Almo hydraulic manifold which is a control panel for hydraulically operated machines are given in 6-page illustrated folder. Internal channels in series of plates carry fluid from power supply through needed controls to strategically located outlets.

54. Tempered Plate Class

Libbey-Owens-Ford Glass Co.—Uniform load strength, resistance to thermal shock and resistance to impact are some of the advantageous properties of Tuf-Flex tempered plate glass which adapt it to use in wide range of equipment. Properties and available sizes are outlined in 4-page bulletin TF-8.

55. Split Roller Bearings

Cooper Split Roller Bearing Corp.—Cooper split roller bearings are made in halves throughout and are readily assembled on the shaft. They are available in bearing, cartridge and pillow block types for medium, heavy and extra heavy duty service and in special designs. Standard sizes range from 1 3/16 to 12-in. bores. Complete details are given in 20-page illustrated design manual.

56. Iron Powder Parts

National Radiator Co., Plastic Metals Div.—28-page illustrated booklet entitled "The Processing, in the Higher Density Range, of Durable Iron Powder Parts on a Production Basis" is a treatise on electrolytic iron powder parts.

57. Corrosionproofing Aluminum

Parker Rust Proof Co.—Described in 4-page illustrated bulletin 3510, Bonderite 710 for aluminum is simple treatment which imparts corrosion resistance to unpainted aluminum and gives increased durability for paint systems.

58. Ball Bearing Practices

Marlin-Rockwell Corp.—Over 5000 additions and revisions have been made in the 92 pages of bulletin No. 26 "Ball Bearing Practices for the Shop Man." Ball bearings of various makes are listed in numerical order and corresponding sizes of M-R-C replacements are given.

59. Gas & Liquid Bellows

Titeflex, Inc. — Text and diagrams found in 6-page bulletin 300 explain the welded diaphragm construction of line of metal expansion bellows for conveying gas and liquids. They are able to withstand high temperature, vibration and corrosion. Specifications are given.

60. Platen-Connecting Hose Units

American Brass Co., American Metal Hose Branch—Bracketubes are seamless flexible metal hose connectors with supports for joining steam-heated platens of presses. Data are given on available styles of units with inside diameters ranging from ⅜ to 1 in. Also covered in 12-page bulletin BR-190R are flexible metal tubing loops in sizes to 3 in. ID and up to 4-in. seamless flexible metal tubing.

61. Electronic Control

Bristol Co.—16-page illustrated bulletin B226 deals with new line of Free-Vane electronic controllers in indicating and recording types for temperature, pressure, flow, liquid level, humidity and time program. Presented are applications and available accessories.

62. Universal Motor Parts

Portable Electric Tools, Inc., Motor Div.—"Universal Motor Parts for Built-In Applications" is title of 6-page illustrated bulletin which describes available facilities for design and production of motor components. Tabulated are dimensions of standard parts for motors with full load outputs of 52 to 400 w at speeds to 15,500 rpm.

63. Roller Bearings

Torrington Co.—Complete design and application data on line of self-aligning spherical roller bearings are contained in 28-page revised bulletin 200-C. Included are width tolerance chart; complete thread, locknut and lock washer data; revised interchangeability chart; and information on life expectancy, capacity ratings, speeds, loads, fits and lubrication.

64. Dripproof Industrial Motors

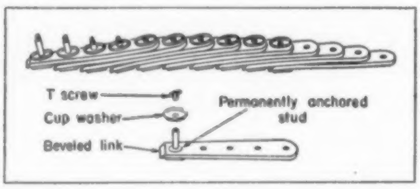
Lima Electric Motor Co.—Specifications, general data and photos of line of dripproof induction motors are found in 6-page bulletin RS-2. Included are standard, vertical, flange and face mounted, magnetic brake, shell type and special motors with ratings from 1/3 to 125 hp for frames from NEMA 66 to 505.

NEW PARTS AND MATERIALS

For additional information on these new developments, see Page 177

Adjustable V-Belt

For D and E drives, Veelos TD and TE link V-belts are made of plys of treated high tensile strength canvas duck, joined by riveted studs with removable cup washers and T-screws. This construction provides strength at key points and facilitates coupling and uncoupling. Belts are available with rubber coating for general services and oil-proof for oily and high temperature drives. Applications include use on large machine tools, presses, shears, compressors, generators, pumps, diesel engines, fans and blowers. Made by **Manheim Mfg. & Belting Co.,** Manheim, Pa.



For more data circle MD-65, Page 177

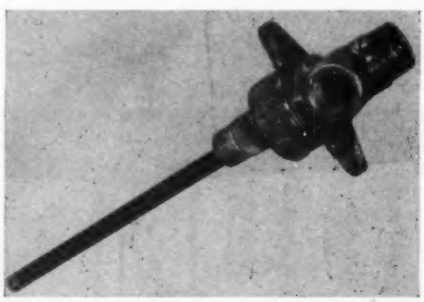
Silicone Rubber

Long term shrinkage of Silastic 675 is only 1.8 to 2.5 per cent. Compression set is very low for a silicone rubber with nontoxic additives—in the range of 15 to 20 per cent after 22 hours at 300 F, 20 to 28 per cent after 70 hours. This material also has superior physical strength and tear resistance in the range of 80 to 90. It is serviceable at temperatures ranging from below −100 to above 500 F. Applications include gaskets, O-rings, seals, diaphragms, bellows, dielectric fittings and connectors, and molded industrial rubber goods. The material can also be molded in dies designed for organic rubber parts if tolerances are not critical. Made by **Dow Corning Corp.,** Midland, Mich.

For more data circle MD-66, Page 177

Double Rotating Joint

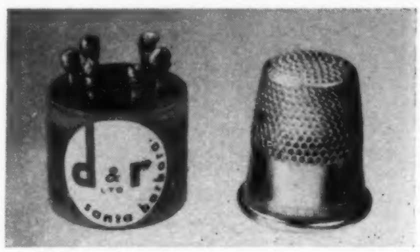
Double passage rotating joint handles fluids at up to 1500 psi. It can be used with both air and hydraulic fluids simultaneously, with members rotating in same or opposite directions as required. It is manufactured in ½ through 5-in. sizes. Among other applications it offers means of connection to double-acting rotating or oscillating air and hydraulic cylinders. Made by **Rotherm Engineering Co. Inc.,** 7280 W. Devon Ave., Chicago 31, Ill.



For more data circle MD-67, Page 177

Subminiature Transformers

Subminiature supply transformers exhibit high performance characteristics in relation to their size and weight. Produced to conform with individual requirements and specifications, they withstand severe shock, temperature ranges and vibration. Model DR-T 107, illustrated, was developed primarily for printed circuit applications. It delivers up to 5 w at 400 cycles per second and is designed to feed a pair of synchro control transformers. It weighs 2/3 oz and is ⅝-in. high and ¾-in. in diameter. Made by **D & R Ltd.,** Santa Barbara, Calif.



For more data circle MD-68, Page 177

Socket Screws

Improved types of points are being applied to Allen set and cap screws. Smaller cup point for set screws, called Allenpoint, replaces the ASA cup point set screws in Allen line. Improvement makes for greater locking at all measured installation vs. removal torque

pressures, uniformly high shaft holding power and good holding power under vibration. This screw is made in Allenoy and stainless

steel, with either National Coarse or National Fine threads. Cap screws now have an unthreaded leader point which reduces causes of screw thread injury and damage to threaded holes. Made by **Allen Mfg. Co.,** 113 Seldon St., Hartford, Conn.

For more data circle MD-69, Page 177

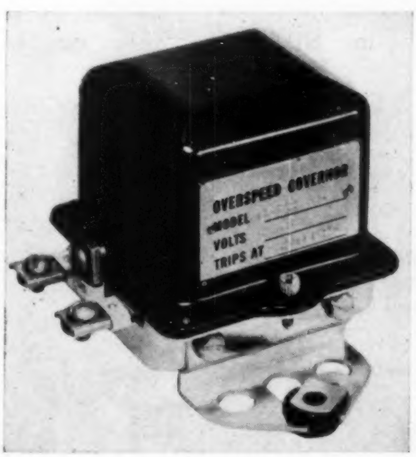
Aluminum Paint

Fast drying aluminum paint for burners, stoves, hot piping and other metal surfaces withstands temperatures up to 1200 F without discoloring or blistering. It dries hard in about 30 minutes and is ready mixed for application by brush or spray. Composed of fine aluminum and a chemical binder, it fuses with the metal as temperatures are increased. Made by **Sapolin Paints Inc.,** 229 E. 42nd St., New York 17, N. Y.

For more data circle MD-70, Page 177

Electric Governors

Frequency sensitive governors do not require mechanical connections to the engine or device to be

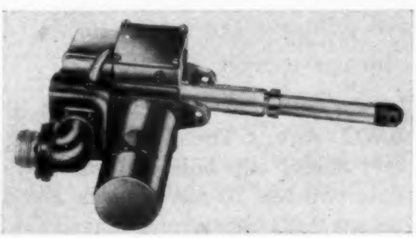


controlled. They can be furnished to receive frequency reference from an ac alternator, either 60 or 400 cycles; from a small tachometer generator; or from the ignition breaker points on a gasoline engine. The frequency sensor consists of a tuned vibrating reed driven by an electromagnet connected to the frequency reference. When this frequency reference resonates with the fundamental mechanical frequency of the reed, the latter vibrates and closes contacts to pull in a control relay. Energy required for operation of the reed is less than 1 w, and the resonant frequency can be calibrated to 1 per cent. Standard models are furnished with single-pole, double-throw contacts and for either automatic or manual reset. Mounting in a space less than 6 in. square, the industrial model weighs less than 2 lb; the aircraft model weighs 11 oz. Made by **Custom Built Controls,** Electro Governor Div., 1801 Rand Rd., Des Plaines, Ill.

For more data circle MD-71, Page 177

Lightweight Actuator

Requiring as little as 2⅜ x 3¾ x 5¼ in. of space and weighing 1¾ lb., Model 396 series linear



actuator makes possible linear movement of loads up to 125 lb. Speed of operation, varying with the gear ratio and the motor utilized, ranges from 1.4 ips at 7 lb load to 0.045 ips at 85 lb load, with the smallest motor available, and from 2.35 to 0.12 ips at the maximum load with the most powerful motor available. It is capable of driving an opposing load of 125 lb through 19½ in. of travel. Included are a gear box; a standard ac or dc motor; a centralizing, modified Acme screw; end position, nonjamming stops; a load limiting device; an internal, slipping-gear type clutch; and a radio noise filter. The load limiting device provides for control of the linear force at any point between 75 and 125 lb, within a fairly narrow band. Made by **Lear Inc.,** Grand Rapids Div., 110 Ionia Ave. N.W., Grand Rapids 2, Mich.

For more data circle MD-72, Page 177

Plastic Tubing

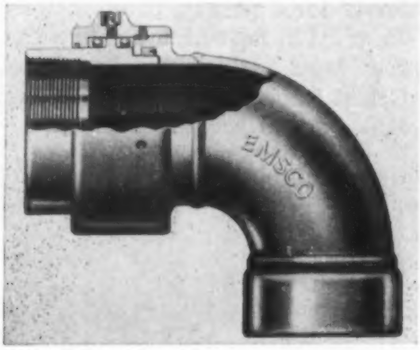
Transparent Ace-Flex plastic tubing for temporary or permanent fluid lines is inert, odorless, nontoxic and withstands effects of aging. It is available in standard sizes with inside diameters ranging from 0.120 to 1 in. It can be made in special formulations to

provide desired combinations of physical, electrical and chemical properties, and special sizes and colors are also available. Standard rubber tube system assembly methods are used. Applications include tubing for coolant systems, lubricating, paint sprayers, air lines, instruments and distilled and food products. Made by **American Hard Rubber Co.,** 93 Worth St., New York 13, N. Y.

For more data circle MD-73, Page 177

Low-Pressure Joint

For use on schedule 40 pipe, type LPR ball bearing swivel fitting is designed for a maximum pressure of 1000 psi at a maximum temperature of 225 F. Characteristics include easy turning at low torque, low resistance to flow, and safe operation. The thrust load is taken directly through the center of the balls. The hardened and ground races on which a double row of ball

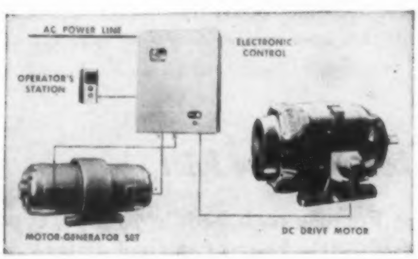


bearings rotate are perpendicular to the lines of force. The fittings handle liquids, gas or semi-solid products which are compatible with the type of synthetic rubber packing used. Packing materials, including Teflon, are independent of the ball rows and are unaffected by ball race wear. Repacking can be done while the fitting is in service. Made in eight basic styles for single, double and triple swings, they may have scarfed, bored, threaded or flanged end connections. Made by **Emsco Mfg. Co.**, Dept. 20, P. O. Box 2098, Terminal Annex, Los Angeles 54, Calif.

For more data circle MD-74, Page 177

Variable-Speed Drives

Electronic control of type GV Speedranger variable speed drive permits fast response to signals based on speed, load, current, voltage, pressure, light, temperature or time. Three or two-phase alternating current power is converted



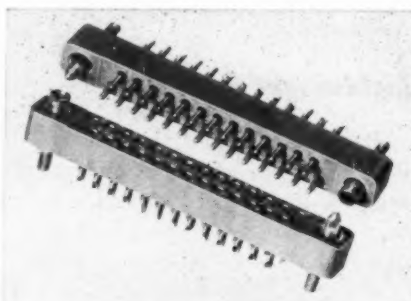
by motor-generator set and electronic tube rectifiers to supply direct current for speed shunt wound motor. Drives are offered in 2 to 10 hp sizes and have basic speeds of 2400, 1750 and 1150 rpm. Drive speeds are adjustable down to one-sixth of basic speed for continuous duty, 50 C, and to one-tenth for intermittent duty.

Line also includes type EV Speedrangers in sizes 1/8 to 1½ hp which are similar in operation except no motor-generator set is used. Optional features for both types are jogging, reversing, dynamic braking and special duty cycles. Direct current drive can be supplied in all standard Master parallel or right-angle shaft gearhead motors with ratios as high as 432 to 1. Made by **Master Electric Co.**, 126-33 Davis Ave., Dayton, O.

For more data circle MD-75, Page 177

Miniature Connector

A contact larger than the No. 20 and smaller than the No. 16 contact used on larger connectors is provided in the series 18 connector



by the use of a contact for No. 18 AWG wire. The 0.053-in. diameter solder cup hole can accommodate two No. 20 AWG wires. Measuring 2 31/32 in. in length, the connector is available with 27 pins. Positive polarization is achieved by the use of reverse type guide pin and guide socket arrangement. Disengagement forces are reduced to a maximum of 8 oz per contact without sacrificing millivolt drop. These connectors can be molded from Melamine for high dielectric and mechanical strength; Plaskon reinforced alkyd type for high impact strength and arc resistance; or diallyl phthalate, mineral or Orlon filled, for high dimensional stability, good dielectric properties and maximum moisture resistance. Breakdown voltage between contacts, with contacts engaged at normal sea level conditions, is 2800 v rms. Pin contact diameter is 0.056-in.; mechanical spacing of contacts is 0.150-in. minimum; effective creepage between contacts is 0.070-in. minimum. Available from **DeJur-Amsco Corp.**, 45-01 Northern Blvd., Long Island City 1, N. Y.

For more data circle MD-76, Page 177

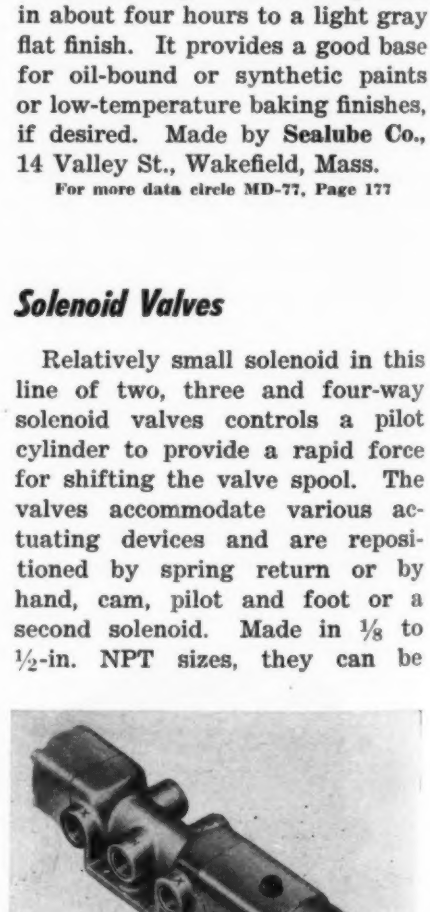
Rust Preventive

ZRC provides a protective coating to iron and steel surfaces, even if applied over adherent rust, mill scale and old paint. The dried coating contains 93 to 95 per cent zinc, which affords electrochemical protection to surfaces. Rust which may form in abrasions in the coating will not spread under it. Recommended method of application is by brushing. The coating dries in about four hours to a light gray flat finish. It provides a good base for oil-bound or synthetic paints or low-temperature baking finishes, if desired. Made by **Sealube Co.**, 14 Valley St., Wakefield, Mass.

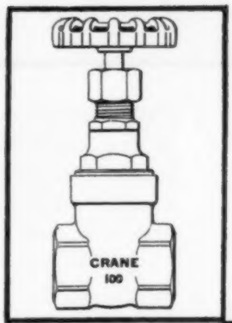
For more data circle MD-77, Page 177

Solenoid Valves

Relatively small solenoid in this line of two, three and four-way solenoid valves controls a pilot cylinder to provide a rapid force for shifting the valve spool. The valves accommodate various actuating devices and are repositioned by spring return or by hand, cam, pilot and foot or a second solenoid. Made in 1/8 to ½-in. NPT sizes, they can be



NEW* brass gate valves



No. 410,
100-Pound

by

CRANE

No. 438,
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No. 437,
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for 3 All-Time CRANE favorites

Here are valves that didn't have to be changed . . . because each has long been the best in its class. But Crane found ways of improvement to give you an even better buy for your brass valve dollars.

For example—here is greater strength, greater rigidity, made possible by the new cylindrical upper body . . . the same basic shape as high-pressure steel valves. Here, too, is an improved stuffing box that screws *into* the bonnet—also better stem support to assure truer alignment and minimize wear on the packing.

And not to be overlooked is their clean, modern appearance—a very desirable advantage for all of your "exposed" installations. Sizes ¼ to 3 in. Ask your Crane Representative all about this improved brass valve line next time he calls.

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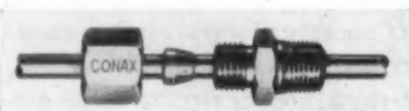
spring centered with numerous arrangements of pressure spools which control the flow. They can be used for pressures up to 150 psi without an additional line, or up to 500 psi with an auxiliary low-pressure line of 35 to 150 psi for supplying the solenoid pilot. Valve's kinetic O-ring seals have a life expectancy of over 3 million cycles. Made by **Versa Products Co. Inc.**, 249 Scholes St., Brooklyn, N. Y.

For more data circle MD-78, Page 177

Tube Fitting

Midlock tube fitting, fabricated of type 303 stainless steel, locks and seals at any location on a tube. Assembly consists of sliding the tube into the fitting and tightening it in the desired position. The ferrule is deformed against the tube and provides a seal without cutting into the tube surface and

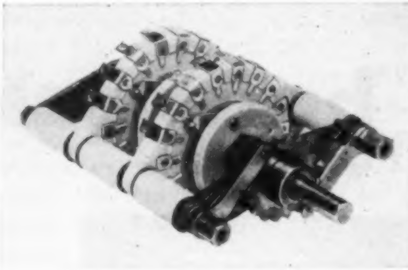


without stress concentration. The fitting can be loosened and resealed repeatedly. Made by **Conax Corp.**, 4515 Main St., Buffalo 21, N. Y.

For more data circle MD-79, Page 177

Selector Switch

The Brown-Hill precision, low-loss, ceramic selector switch is used in applications requiring constant stability in conductivity and minimum resistance, inductance and capacitance. The 18-position wafer type switches have a voltage breakdown rating of 4000 v ac, peak, at 60 cycles. Current carrying capacity is 30 amp at 60 cycles, noninductive load. It has a 20-degree positive detent mechanism with adjustable stops. Rotor blades and stator contacts are of solid silver, providing low resistance, high current capacity and eliminating receiver switching

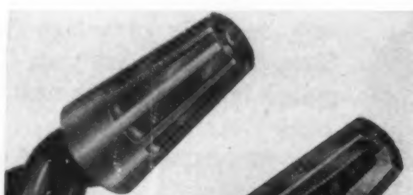


noise. Hole and cutouts are provided in the contact for mechanical attachment of wires. Made of Dow Corning-200 impregnated steatite throughout. Available in one to six gangs and either bushing or stud-mounted from **R-F Electronics Inc.**, 291 N. E. 61st St., Miami, Fla.

For more data circle MD-80, Page 177

Plastic Wire Connectors

Screw-on wire connectors with one-piece molded plastic threads and body facilitate connection of No. 14, 16 and 18 wires. Made in two sizes, connectors have high impact strength and good dielectric properties. Smooth threads grip wires securely without cutting wire strands. They are green in color, withstand temperatures up to 400 F and bear Underwriters' approval.



Made by **Ideal Industries Inc.**, 1059 Park Ave., Sycamore, Ill.

For more data circle MD-81, Page 177

Voltage Supply

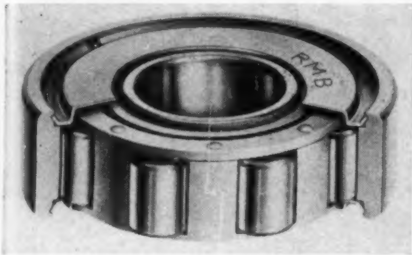
Constant reference voltage supply converts alternating current into a direct current voltage which remains constant within $\frac{1}{8}$ of 1 per cent. Designed primarily as a reference voltage for variable-speed motor drives and dc servo systems, it can also be used to replace standard cells, wet or dry cells, or electronic regulating systems. When connected to a 115-v supply of 50 to 400 cycles, the unit supplies a reference potential adjustable from 0 to approximately 87 v. This reference voltage is maintained with any load not exceeding 1 ma and throughout a temperature range of -50 to 40 C, with line voltage

changes from 90 to 135 v. Two models are available, supplying, for external use, 275 volts unregulated, not exceeding 5 ma; or 150 ± 2 volts regulated, not exceeding 5 ma. Both supply a filament voltage of 6.3 v ac, with center-tap, at a current not exceeding 0.6-amp. Made by **Servo-Tek Products Co. Inc.**, 1086 Goffle Rd., Hawthorne, N. J.

For more data circle MD-82, Page 177

Miniature Roller Bearings

Three new types of RMB miniature roller bearings are suitable for use in apparatus, machines and motors where shafts are subjected to heavy radial loads. They permit some axial displacement of the shaft, thus making possible more rapid assembly. Seven sizes have outside diameters from 0.4724 to 1.0236-in. Roller assembly is retained by a snapring on the inner race and section of the shield at the outer race. Type N permits free axial displacement of the



shaft and easy removal of shaft, inner race and roller cage as a unit. In type NU the inner race remains on the shaft, while outer race, roller and cage can be removed as a unit. Type NP (illustrated) has permanently assembled inner and outer races. Available from **Landis & Gyr Inc.**, 45 W. 45th St., New York 36, N. Y.

For more data circle MD-83, Page 177

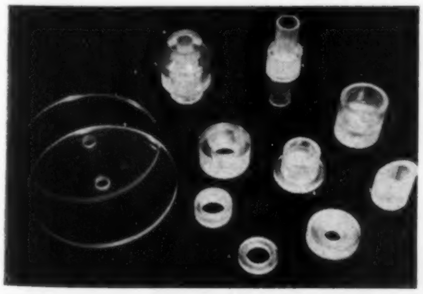
Lubricating Grease

An oxidation-inhibited lithium base product, Molykote BR2 is a multipurpose extreme pressure lubricating grease containing molybdenum disulphide. It is useful for highly loaded ball and roller bearings and sliding friction surfaces. Operating range is from −30 to 350 F. Made by **Alpha Corp.**, 179 Hamilton Ave., Greenwich, Conn.

For more data circle MD-84, Page 177

Plastic Insulating Material

Easily machined and possessing a low dissipation factor, Stycast 0005 insulating material will with-

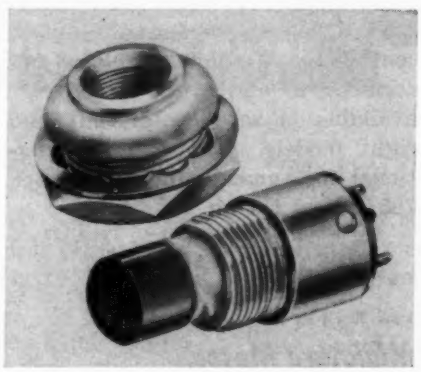


stand temperatures as low as −70 C and will not flow when subjected to temperatures as high as 200 C. Prolonged heating under stress, however, should not exceed 125 C. Available in rods in nominal diameters from 1/4 to 3 in. and in sheets in nominal thicknesses from 1/4 to 1 in., the material has tensile strength of 11,000 psi and a Rockwell M scale hardness of 105. At frequencies of 60 to 10^{10} cycles per second, the dielectric constant is 2.53 to 2.56; dissipation factor is below 0.0005. Made by **Emerson & Cuming Inc.**, 869 Washington St., Canton, Mass.

For more data circle MD-85, Page 177

Pushbutton Switches

Series W100 snap-action switches are made to meet MIL-S-6743 specifications and incorporate a moisture-proofing silicone rubber sleeve. Available types include most commonly used SP-ST, 3 terminal or SP-DT circuit arrangements. Current ratings are 10 amp resistive, 5 amp inductive and 3 amp lamp. Units are fabricat-

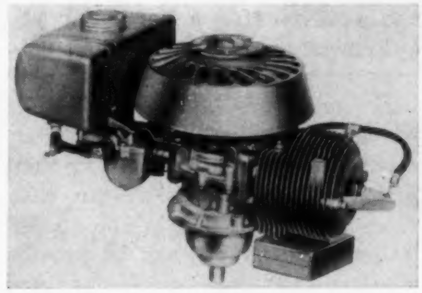


ed from lightweight anodized aluminum, and adaptors are available to meet nearly all mounting requirements. They are 1 5/16-in. long and weigh approximately 1/4 oz. Made by **Hetherington Inc.**, Sharon Hill, Pa.

For more data circle MD-86, Page 177

Gasoline Engines

Lightweight models AV 47 and AH 47 have concentric cylinder fins. Both models develop 2 hp, have 2-in. bore, 1½-in. stroke, 4.7



cu in. displacement, and horizontal cylinders. The crankshaft is vertical on model AV 47, horizontal on model AH 47. Vertical model weighs 13 lb and is 14½ in. long; horizontal model weighs 14¼ lb and is 13 in. long. Made by **Power Products Corp.**, 1029 13th Ave., Grafton, Wis.

For more data circle MD-87, Page 177

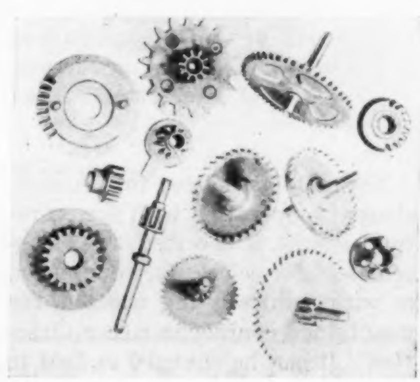
Dual Wheel Caster

Shock-absorbing Aerol wheel and caster assembly has individually sprung dual 16-in. wheels, giving smooth load travel over rough surfaces. It is fitted with an anti-shimmy brake for high speed operation. It weighs 75 lb, and is cast from heat treated 365 aluminum. Made by **Aerol Co.**, 1731 Workman St., Los Angeles 31, Calif.

For more data circle MD-88, Page 177

Small Gears and Pinions

Small gears and pinions are die cast in one piece from large stock of interchangeable die elements. Tooling combinations can be assembled from a large variety of gear and pinion or hub dies. Tools



not available for a particular requirement can be made when volume warrants. Small gears, pinions and gears with combination of mechanical elements such as cams, hubs, spacers and flanges are available, precision die cast from zinc alloy. Shafts and center holes of all shapes can be provided. Maximum gear dimensions are 1 5/16 in. OD and 1/16-in. face width. Made by **Gries Reproducer Corp.**, 400 Beechwood Ave., New Rochelle, N. Y.

For more data circle MD-89, Page 177

Snap-Action Limit Switch

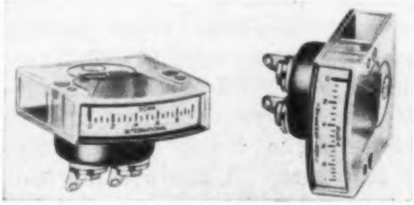
Features of Loxswitch single-pole double-throw limit switch include a trigger-action break independent of operating lever speed, operating shaft supported by bearings at both ends and large wiring compartment separated from moving parts. Vibration and impact shock are minimized by use of lightweight silver contacts in balanced rotor. Each pole is independently spring loaded and locked when in either position.

Normally assembled for counterclockwise operation with spring return, switch, if specified, can be assembled for clockwise operation, or with return spring omitted, for maintained contact in either direction. It can be changed in field to meet these conditions. Usable on up to 600-v circuits, switch is rated 6 amp on 120-v ac. With 20-degree over-travel, units have tested at 20 million operations. Made by **R. B. Denison Mfg. Co.**, 4220 Hough Ave., Cleveland 3, O.

For more data circle MD-90, Page 177

Panel Meters

Line of miniature side indicating panel meters provides maximum scale length with minimum panel area. Available in various ranges with flanges for single and back-to-back mounting, the self-contained units can be grouped both horizontally and vertically. The meters are accurate to ± 3 per cent of full

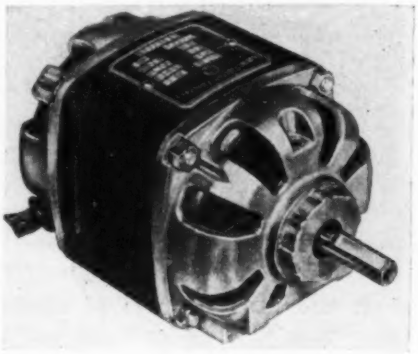


scale deflection on dc and ± 5 per cent on ac. Ammeters, voltmeters, decibel, vu, and other meters are available in zero center, left and right models. Made by **International Instruments Inc.**, P. O. Box 2954, New Haven 15, Conn.

For more data circle MD-91, Page 177

Miniature Motor

Height and width of only 2⅞ in., constant speed, quiet operation, no radio interference, light weight and five mounting arrangements are characteristics of type K-4 induction motor. It is rectangular and has end shields for flush and recessed base, resilient ring, extended stud end and semi-special flange mountings. Motor is made in three sizes as either a split phase or capacitor type with constant speed of 1725 or 3450 rpm. Applications include use in typewriters, magnetic tape recorders, movie projectors and automatic phonographs. Made by

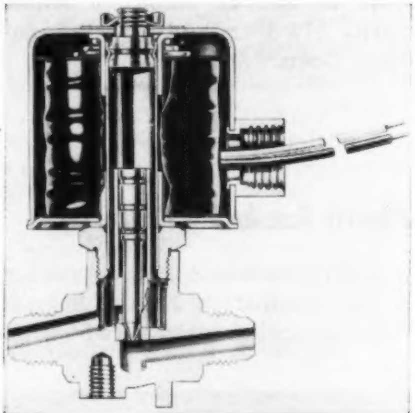


Bodine Electric Co., 2254 W. Ohio St., Chicago 12, Ill.

For more data circle MD-92, Page 177

Solenoid Valve

Three orifice sizes, 5/32, 3/16 and 7/32-in., and four body styles are available in improved model 73 solenoid valve. Double-dipped varnish impregnated coils which withstand high humidity conditions are easily changed to suit voltages required for special applications. An electrical data plate accompanies each interchangeable coil. The valve can be used with any noncorrosive liquid, refriger-



ants or air. Made by **A-P Controls Corp.**, 2450 N. 32nd St., Milwaukee 45, Wis.

For more data circle MD-93, Page 177

Powder Metal Alloy

Two grades of Hevimet, an alloy of tungsten, nickel and copper which is 50 per cent heavier than

lead, are available for special applications. HM-1, with high tensile strength, provides greater rotary stabilizing force in applications such as gyro rotors. HM-2 affords 1½ times more gamma ray protection than lead and is adapted to radiation shielding. Both grades are machinable with carbide cutting tools and can be cadmium, chromium or nickel-plated. Made by Carboloy Dept., **General Electric Co.,** Detroit 32, Mich.

For more data circle MD-94, Page 177

Thermal Sensing Unit

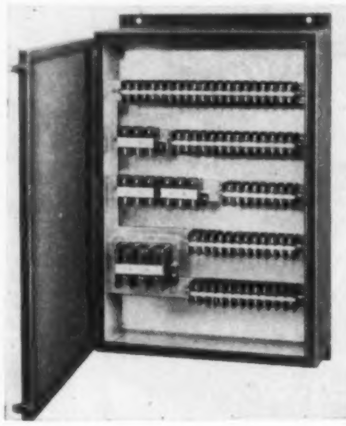
This thermal sensing unit, called Plugstat, provides precise thermal warning without calibration drift. It can be used wherever temperature limit indication is required, as

in aircraft generators, air and water cooled engines, bearings, etc. Alloy contacts give clean make-and-break. Unit is capable of a 200 F overshoot from a high or low normal setting. It is rated at ¼-amp, 28 v, ac or dc resistive loads. Contacts can be arranged normally-open or normally-closed. Instrument is factory set and sealed and weighs approximately 1½ oz. Made by **Control Products Inc.,** 300 Sussex St.. Harrison, N. J.

For more data circle MD-95, Page 177

Terminal Junction Boxes

Largest unit in line of eight standard terminal junction boxes will accommodate 168 25-amp terminals. Fifty and 100-amp terminals can also be used in combination with the 25-amp terminals. External mounting feet are provided, and hinged door is gasketed with Neoprene to prevent entrance of oil or coolants when used on machine tools. Terminals are mounted on removable straps and have generous wiring space. With

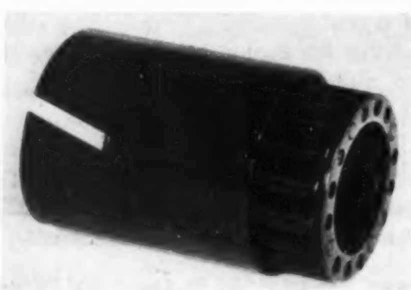


no knock-outs, box can be entered as desired. Made by **Square D Co.,** 4041 N. Richards St., Milwaukee 12, Wis.

For more data circle MD-96, Page 177

Roller Bearing

Steel cage solid cylindrical roller bearing is designed for use on a hardened and ground shaft where space is limited and the housing cannot be hardened and ground to required tolerances. Designated type A, inch series, the bearing has a roller assembly and split outer sleeves; no inner race is necessary. The roller assembly is formed by solid, hardened steel alloy rollers mounted to two hard-

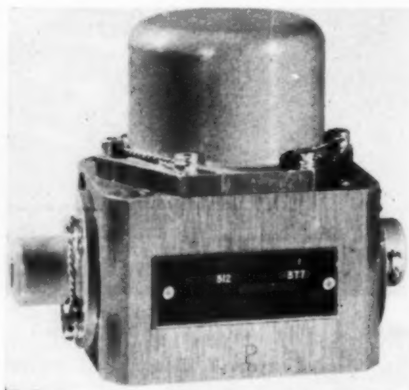


ened end rings. The split outer sleeve forms an obtuse angle joint; rollers pass over the joint without shock or noise. Hardened steel construction permits heavy duty performance at speeds to 1000 rpm. Parts of the bearing can be installed separately into the machine in which it is to be used, being joined when the machine is assembled. Made by **Rollway Bearing Co. Inc.,** Seymour St., Syracuse, N. Y.

For more data circle MD-97, Page 177

Servo Valve

Any maximum desired output flow from 0.5 to 8.0 gpm in 1000 to 3000 psi hydraulic systems can be controlled by the model 500 electro-hydraulic servo valve. Standard signal input for full output flow is a differential current of ±8.0 milliamperes, with lower or higher signal levels also avail-



able. Use is in control systems requiring high accuracy or high speed of response. Valve body is aluminum alloy, and bushing and spool assembly are hardened steel. Size is 2 7/16 in. high by 3 1/16 in. wide by 1¾ in. deep. Made by **Moog Valve Co. Inc.,** Proner Airport, East Aurora, N. Y.

For more data circle MD-98, Page 177

Heat Resistant Enamel

Kemclad Hi-Heat enamel for coating household appliances and other equipment exposed to sustained, moderately high temperatures is a combination of inorganic and organic materials. A modification of the silicone class of materials, it retains the heat resist-

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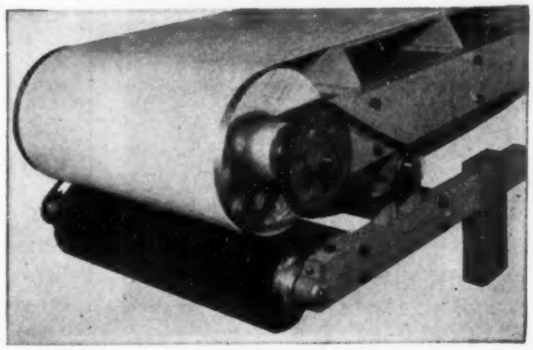
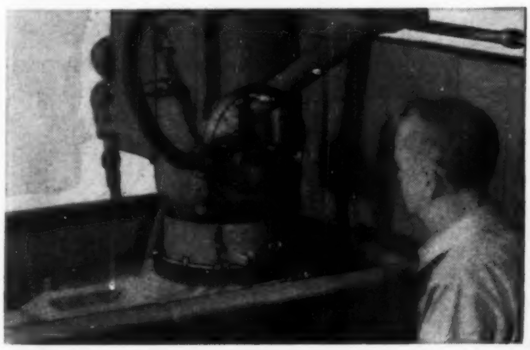
This brush strip can be bent, coiled, or twisted into numerous shapes. It can be formed to give intermittent or continuous brushing action. It can be attached to stationary or power driven mountings . . . can be adapted to your present machines or designed for new machine uses. Size possibilities are almost unlimited. To date they run from ¼ inch to 18½ feet in length. How it may help *you* is a matter of your own ingenuity — plus the services of the Fuller Brush Engineering Dept. Find out what Fullergript can do by sending for a sample strip. We will also send a booklet showing its versatility. Simply write us.



Power-driven Fullergript Brush, mounted against a conveyor belt where it passes around the end pulley, cleans off all material . . . saves 14 man-hours formerly spent removing carry-back.



Fullergript brush strip mounted on the guard surrounding the grinding wheel of a vertical die and surface grinder. The Fullergript strip concentrates the coolant on the work . . . and away from the operator.

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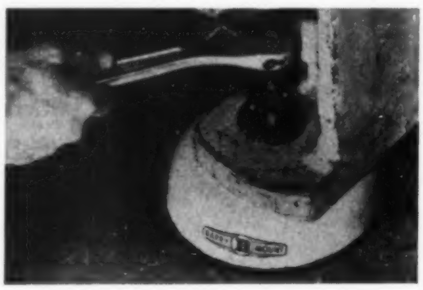
ant properties of inorganics and possesses the versatility and economy of an organic finish. Nonyellowing under sustained temperatures up to 500 F, it retains its glossy finish despite exposure to high temperatures and resists chipping and the effects of salt spray, hot grease, humidity, chemicals and household cleansers. Currently offered in white only, the enamel may be produced in a range of colors later. Developed by **Sherwin-Williams Co.,** Cleveland 1, O.

For more data circle MD-99, Page 177

Vibration Isolators

Turning the attaching bolts of series LM3 and LM5 Barrymounts lifts a machine foot to the required height for leveling. Horizontal, vertical and rotational vibration is isolated. Height adjust-

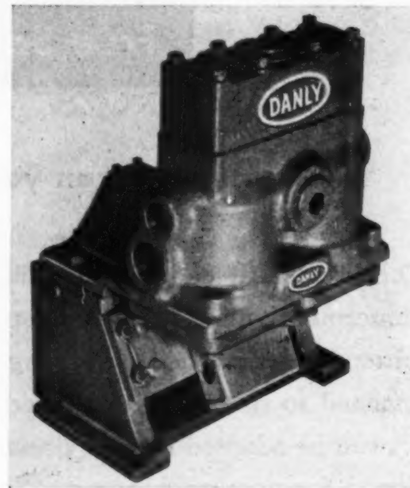


ments up to ½-in. are possible, and the load limit is 4200 lb per isolator. Made by **Barry Corp.,** 1000 Pleasant St., Watertown, Mass.

For more data circle MD-100, Page 177

Clutch Air Valve

Designed specifically to eliminate accidental stroking due to valve failure, this dual air valve consists of two independent pilot operated valves which are actuated by electromagnets. The main valves are air actuated. Pilot valve construction reduces inrush and holding current requirements. The two independent valves are built into a single unit and cross ported in such a way as to provide maximum safety. Each valve in the unit is a two-position, normally closed type. Constructed for a pressure rating of 125 psi, the unit does not require lubrication



and has no seals or springs. It operates on 115-v, 50 or 60-cycle current. Made by **Danly Machine Specialties Inc.,** 2100 S. Laramie Ave., Chicago 50, Ill.

For more data circle MD-101, Page 177

Electronic Relay

Using an electro-strictive capacitive element, Capaswitch electronic relay requires only 0.5 milliwatt-seconds of operating power to close the contacts, and less than 0.1-milliwatt to hold them closed.

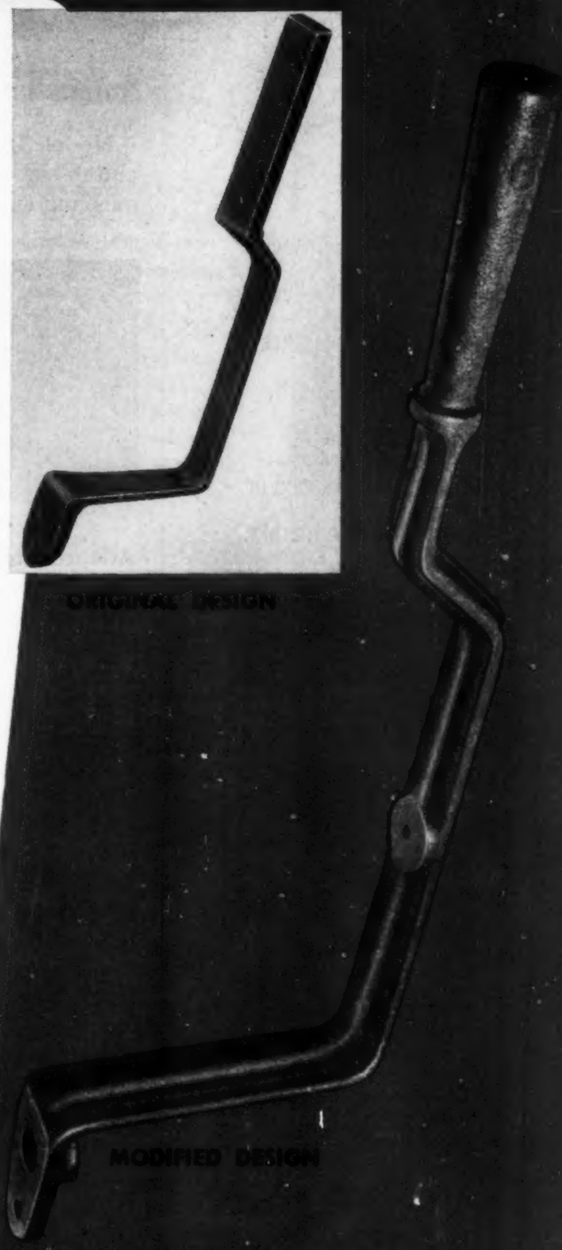
Low power requirement permits direct operation by a phototube without an amplifier. Input resistance of electrostatic element is approximately 100 megohms, and unit is actuated by electrical pulses as short as 10 microseconds. Model A is single-pole double-throw type rated at 1 amp, 110-v ac non-

Case History M-122

Here's what we mean by SUPERIOR ENGINEERED FOUNDRY PRODUCTS...

PROBLEM:

1. This crawler tractor steering lever, a weldment of steel plate and bar stock, was expensive to make.
2. It lacked eye-appeal and detracted from the efficient appearance of the tractor.
3. Dimensional stability was difficult to maintain.
4. Assembling and machining time was excessive.



SOLUTION:

A *SUPERIOR* ENGINEERED FOUNDRY DESIGN MALLEABLE IRON CASTING. It would cost less to produce, it would look better, dimensional stability would be maintained. It would have the rigidity and strength inherent in properly designed malleable iron castings.

RESULT:

1. Production costs reduced.
2. Greatly improved appearance.
3. Dimensional stability easy to maintain.
4. Assembly time eliminated, machining time reduced to a minimum.

TOTAL COST OF PART REDUCED 19.5%.

YOU, TOO, CAN BENEFIT BY CONSULTING *SUPERIOR'S* PRODUCT DEVELOPMENT SERVICE.

It will help solve your design problems, help save you time and money. Through *Superior* Engineered Foundry Design you get maximum advantages of quality, cost, performance, machining, assembly, weight, appearance . . . any or all in a single product.

Make your parts *Superior* Engineered Foundry Products . . . malleable iron castings to 300 pounds, steel castings to 30,000 pounds. It will pay you to get in touch with *Superior*.

Have you entered the first annual Product Development Contest sponsored by the Steel Founders Society of America? We'll be happy to send you the details. Just drop us a line.

SALES OFFICES:
CHICAGO 4, Railroad & Industrial Products Co., 332 S. Michigan Ave.
DETROIT 35, Ray T. Morris, 18050 James Couzens Highway
BUFFALO 21, Gibney-Coffman Corp., 2107 Kensington Ave.

ENGINEERED FOUNDRY PRODUCTS

SUPERIOR STEEL AND MALLEABLE CASTINGS CO.

BENTON HARBOR, MICHIGAN, U. S. A.

Making Good Castings For Quality-Conscious People Since 1916

inductive load. Made by **Mullenbach Electrical Mfg. Co.,** 2300 E. 27th St., Los Angeles 58, Calif.

For more data circle MD-102, Page 177

Automatic Clutch

Model 18C20 V-Plex automatic clutch is designed for use with gasoline engines of 1-in. shaft size.

Adapted to applications requiring high starting torque, it tightens the belt as the throttle is opened. Engagement speed ranges from 1800 to 2500 rpm, depending upon the belt load; disengagement occurs at 1200 rpm. Engagement is automatic as the throttle is opened. As shaft speed increases, driving belt pitch diameter is also increased. Using a B size V-belt, the unit has an overall pulley diameter of 2⅞ in. Installation is accomplished by slipping the clutch over the engine driveshaft and tightening a self locking setscrew. Made by V-Plex Clutch Div., **Light Inspection Car Works,** Hagerstown, Ind.

For more data circle MD-103, Page 177

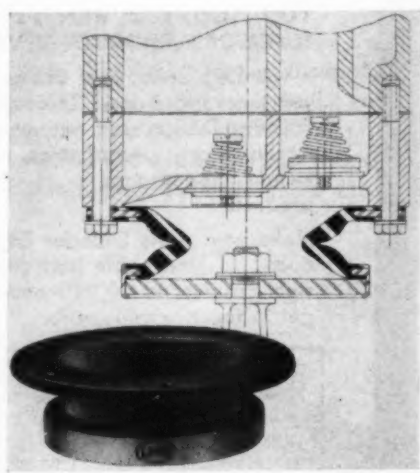
High-Temperature Wire

Temprene stranded copper, silver plated hook-up and lead-in wire is supplied in 11 solid colors and more than 100 combinations of striped insulation. Standard sizes range from 10 to 28 AWG, in both thin and heavy wall thicknesses. Insulated with Teflon, the wire is capable of continuous duty at −100 to 260 C without deterioration and remains unaffected by outdoor weathering. Electrical and chemical properties are excellent over a range of frequencies and temperatures. Dielectric strength is 1000 v per mil, and volume resistivity is 10^9 megohms per cm. The dielectric constant is 2.0 to 2.05 at frequencies from 60 cycles per second to 30,000 megacycles. More than 20 standard combinations of stranding are available, depending on the gage required; other strandings and special wall thicknesses can be supplied. Made by **Hitemp Wires Inc.,** 26 Windsor Rd., Mineola, L. I., N. Y.

For more data circle MD-104, Page 177

Pump Diaphragm

Elastomers which resist corrosive solutions and abrasive slurries are used in the fabrication of this bonded-rubber pump diaphragm. It is claimed that pumps using the diaphragm operate quietly and more efficiently than comparable size piston or jet types. The diaphragm illustrated operated continuously for 3500 hours without a decrease in pump efficiency. Applications include use on hydraulic or pneumatic units. Made



by **Lord Mfg. Co.,** West 12th St., Erie, Pa.

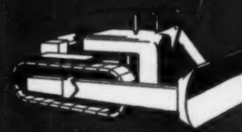
For more data circle MD-105, Page 177

Stainless Steel Valve

Union bonnet, inside screw, nonrising stem stainless steel valve is available in ½, ¾ and 1-in. sizes. It has a compact shell cast body

BEFORE YOU DECIDE ON ANY HYDRAULIC PUMP FOR YOUR PRODUCT LOOK OVER THESE

CHECK POINTS OF *Pesco* SUPERIORITY



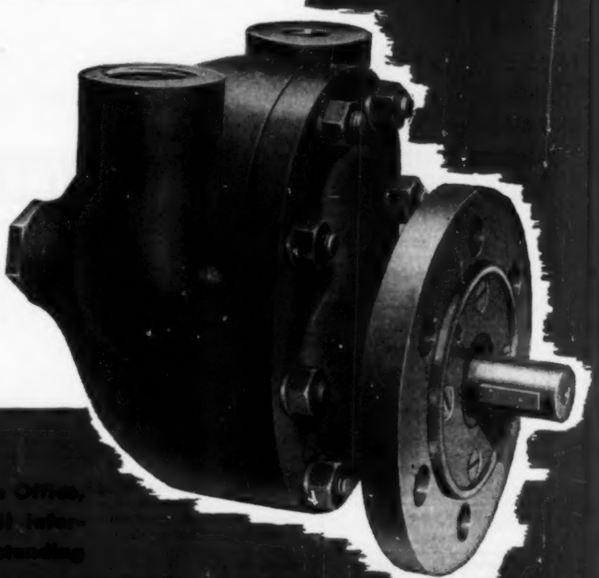
DEPENDABLE PERFORMANCE—Thousands of PESCO Pressure Loaded PUMPS on critical installations, have proved their dependability of operation with exceptional high-efficiency performance of their specific jobs.

CONTINUOUS "NEW PUMP" EFFICIENCY—PESCO quality control, from original design through precision manufacture and final testing, builds into PESCO products every characteristic that will insure life-long, peak-operating efficiency and "new pump" performance—with volumetric efficiencies to 97% and torque efficiencies to 92%.

FULL RANGE OF CAPACITIES AND PRESSURES—PESCO PUMPS provide a full range of capacities to 60 gpm and pressures to 3000 psi. Operating speeds range from 1000 rpm to 4500.

HYDRAULIC ENGINEERING SERVICE—This outstanding PESCO service is available for solution of hydraulic problems. Simply call or write the Home Office, Bedford, Ohio—The combined experience and skill of Pesco Hydraulic engineers is yours without obligation.

LESS UNIT SPACE REQUIRED—The "Pressure Loading" features of PESCO PUMPS reduce the overall size requirement for a given installation. This is possible because PESCO PUMPS can be operated safely at higher speeds—and still maintain the same excellent volumetric efficiencies at slow operating speeds.



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BORG-WARNER CORPORATION

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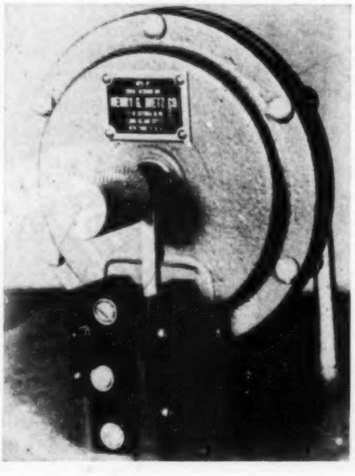
and guided precision cast disk. Most exterior surfaces, with the exception of the cast body, are polished, providing a clean and dirt resistant surface. The nonrising stem minimizes product contamination, while a large stuffing box

and accurately machined disks and seats minimize the possibility of leakage. Heavy wall sections resist corrosion. Made by **Cooper Alloy Foundry Co.,** Bloy St. and Ramsey Ave., Hillside, N. J.

For more data circle MD-106, Page 177

Pressure Controls

Improved Cat. 104 sensitive static pressure controls employ molded screw type terminals. Units are available with silicone impregnated glass cloth diaphragms instead of



No sir . . . They're OK hardened bedway cross sections!

These hardened ways, made by The Ohio Knife Co., give you an idea of the diversified shapes and sizes supplied. OK's special process of manufacture permits building to your specifications in a wide range of design.

Specially heat-treated tool steel makes OK ways practically wear-proof and helps maintain constant accuracy throughout the life of the machine. This extreme hardness, 65-66 Rockwell "C," is uniform to 3/16-inch depth or more.

Through the Ohio Knife process the long-wearing tool steel is welded to a soft steel backing under 2500 tons pressure. Then precision equipment finishes the bedway by grinding to tolerances up to ± .0002 inches.

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neoprene where operation at high or low temperatures over long periods of time is desired. The controls are used on very low pressures where regulation is required in inches of water. They can be made to operate at any value from 0.3 to 20 in. water pressure with fixed differentials of approximately 10 per cent of the operating pressure. Electrical ratings are approved by Underwriters' Laboratories for 10 amp, 125 v or 5 amp, 250 v ac. Made by Coral Designs Div., **Henry G. Dietz Co.,** 12-16 Astoria Blvd., Long Island City 2, N. Y.

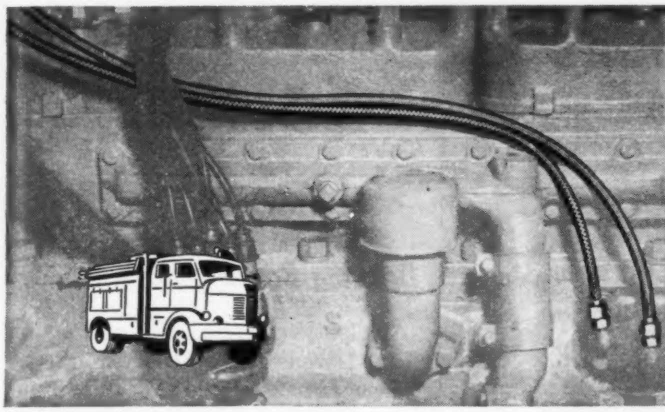
For more data circle MD-107, Page 177

Time Delay Relay

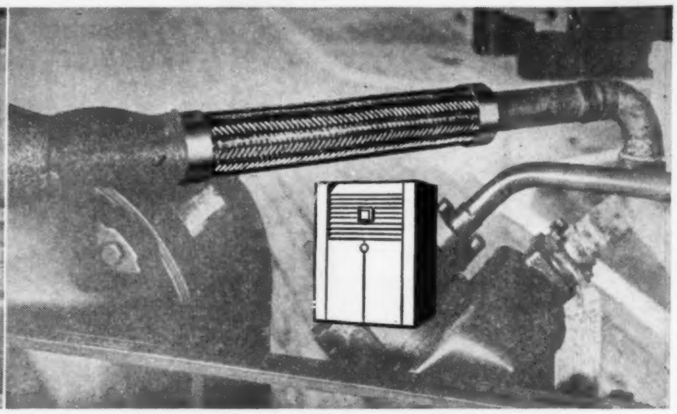
Pushbutton or Microswitch providing a momentary impulse will initiate the time delay of the model NEH Agastat time delay relay. Lightweight compact unit has a pneumatically controlled time delay adjustable from 0.1-second to 10 minutes or more. It has a normally-closed auxiliary hold-in switch. Entire unit is 5⅞ in. high and 2½ in. square. It is available for all standard alternating and direct current voltages in single-pole, double - throw or double-pole, double-throw models. Rating is 15 amp at 115 v 60 cycles ac. Made by A'G'A Div., **Elastic Stop Nut Corp. of America,** 1027 Newark Ave., Elizabeth, N. J.

For more data circle MD-108, Page 177

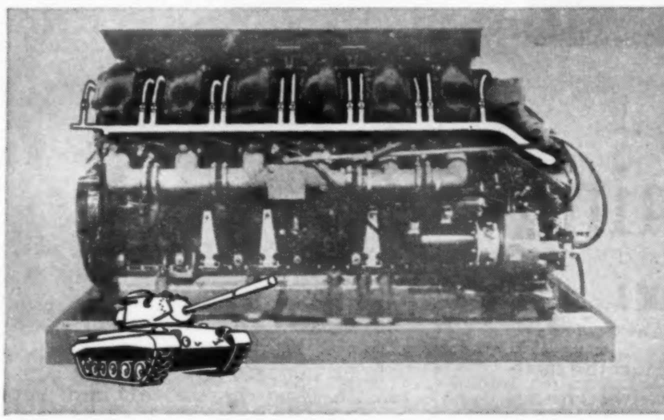
These 4 may end your design worries, too



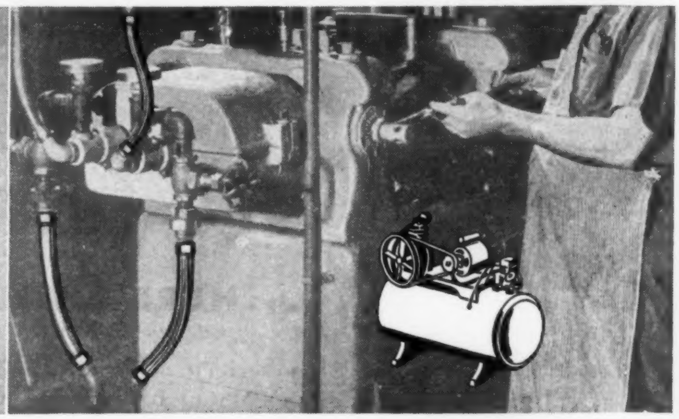
END FUEL LINE FAILURE. Four Wheel Drive's famous trucks stand up to roughest service. So do their Titeflex® oil and fuel lines. Automotive engineers specify Titeflex metal hose because it resists wear, vibration and corrosion —won't crack, bake or deteriorate under high engine temperatures.



ELIMINATE VIBRATION. To end vibration and prevent leaks around fittings, Uniflex seamless metal hose is installed between circulator coils and motor of GE's packaged air-conditioning unit. Made by Titeflex, Inc., Uniflex withstands critical stress and strain—is inherently leakproof.



BANISH IGNITION TROUBLE. Neither mud nor dust, snow, water or lubricants can affect the Titeflex-protected ignition leads of Continental engines in Patton M-48 Tanks. Also suppresses radio interference. Titeflex is a leader in developing ignition harnesses and metal hose lines for Army Ordnance tanks and vehicles. Titeflex quality pays off here.



CONVEY CRITICAL FLUIDS SAFELY. Cooling lines to rubber milling machine use flexibility of Titeflex to advantage. In other applications, tough, corrosion-and-wear-resistant Titeflex safely conveys oil, steam, gases, lubricants, brine, acids, oxygen and compressed air. Rugged, seamless Uniflex withstands extreme vibration, physical abuse and strain.

THERE'S ALMOST NO END to the engineering uses for Titeflex® seamed flexible metal hose or Uniflex seamless metal hose. From aircraft to automotive equipment—from drain lines to dental units—Titeflex simplifies design, construction, operation and maintenance. For types of hose, fittings, assemblies, applications and engineering data, keep our *new 48-page Metal Hose Catalog* No. 200 at your elbow. Use the coupon below to bring it and our design service without delay.

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Titeflex

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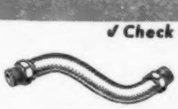

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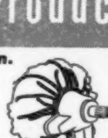
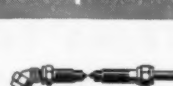
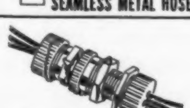
☐ PRECISION BELLOWS



☐ IGNITION HARNESS



☐ IGNITION SHIELDING



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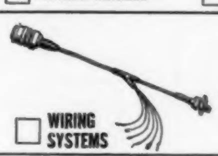
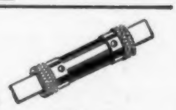


☐ WIRING SYSTEMS



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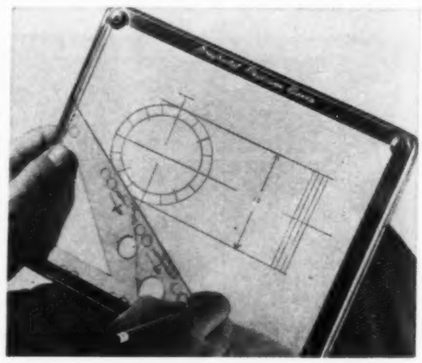
ENGINEERING DEPARTMENT

EQUIPMENT

Drawing Board

Polystyrene plastic drawing board weighs less than 8 oz and measures 9¾ x 12¼ in. Four corner clamps for attaching 8½ x 11-in. paper are recessed into the plastic, and horizontal and vertical retractable metal straightedges are provided. Tension clamps on the under side of the board secure triangles for storage. Made by **Graphostat Co.,** 110 Eaton Place, East Orange, N. J.

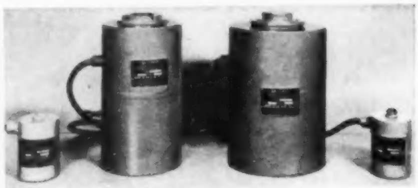


For more data circle MD-109, Page 177

Load Cells

Two series of type SR-4 load cells for measuring forces and weights include 18 compression cells, Type CX and CXX, in nine capacities ranging from 500 to 200,000 lb, and 16 tension load cells, Type TX and TXX, in eight capacities ranging from 500 to 100,000 lb. Nominal resistance of the cells is 120 ohm, and their voltage output at rated capacity is 2 mv per volt. Type CXX and TXX "extra precision" cells have a calibration accuracy within ±0.10 per cent at full-scale output at 65 F. Temperature effect between 15 and 115 F on sensitivity

(output) is within ±0.0008 per cent per deg F; temperature effect on zero load, within ± 0.0015 per cent of full scale per deg F. Zero



balance at 65 F is within ±2.5 per cent of full scale. Calibration accuracy of "precision" cells is within ±0.15 per cent at full scale output. Temperature effect between 15 and 115 F on sensitivity (output) is within ±0.002 per cent per deg F, and the temperature effect on zero load is within ±0.0015 per cent. Compression cells range from 4 13/16 to 11⅝ in. high and 3 to 8 in. in diameter; tension cells range from 4 7/16 to 17¾ in. high and 3 to 9 in. in diameter. Made by **Baldwin-Lima-Hamilton Corp.,** Philadelphia 42, Pa.

For more data circle MD-110, Page 177

Record Reader

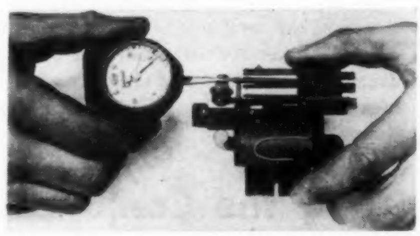
Designed to facilitate measurements from oscillographic or film tracings, the Contact Telereader has an 18 x 30-in. illuminated viewing area. Behind the screen, two crosswires are suspended between separate carriers which, controlled by handwheels, travel on linear ball bushings along precision rails. A cable and pulley arrangement interconnects the carriers and keeps the crosswires perpendicular throughout the viewing area. The position of the crosswires coincides

with the calibrated dials at the right of the operator. These dials are equipped with magnifiers and a light source independent of the incandescent lamp which illuminates the screen. Each positively and negatively marked, the dials are calibrated in hundredths of inches and may be set to zero for any arbitrary crosswire position. Accuracy is ±0.005-in. with visual interpolation. Made by **Telecomputing Corp.,** 133 E. Santa Anita Ave., Burbank, Calif.

For more data circle MD-111, Page 177

Dynamometer

Precision instrument measures spring tension, starting torque and the force required to actuate delicate mechanisms. It is used to determine and check the force required to overcome spring tension and other types of resistance in mechanisms such as electric con-



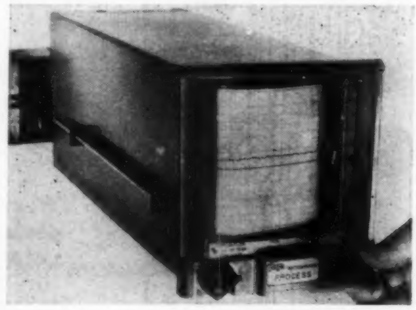
tacts, relays, telephones, clocks, business machines and time switches. A small model is available in pressure ranges of 5 to 15, 5 to 30, 10 to 50, 20 to 100 and 25 to 150 grams; a large model, in ranges of 25 to 250, 50 to 100 and 100 to 1000 grams. All act in both directions and have friction leader hands which, moved by the indicator hands, stop and remain set at the highest force exerted during the test. Made by **George Scherr Co. Inc.,** 200 Lafayette St., New York 12, N. Y.

For more data circle MD-112, Page 177

Miniature Recorder

Capable of receiving four variables, the Ratographic recorder can record two variables and in-

dicate two more or record three variables and indicate a fourth. A field-mounted pneumatic controller plugged into the back of the recorder chassis transforms the recorder into a recording-controller. Suitable for standard, graphic or



semi-graphic panel mounting, the instrument requires 6 x 6 in. of panel space and uses a 4-in. rectilinear strip chart available with either electric or pneumatic drive. Made by **Fischer & Porter Co.,** 13 Jacksonville Rd., Hatboro, Pa.

For more data circle MD-113, Page 177

Sine Wave Generator

For the determination of transfer functions of automatic control systems components and for defining the physical properties of existing plants to which control systems are to be added, this instrument acquires data by the frequency synthesis method. Sinusoidal forcing functions are applied to the system and measurements taken of amplitude and phase shift of the output response. Adaptable to pneumatic devices prevalent in industrial control systems, it generates sinusoidal air

pulses of adjustable frequency, amplitude and mean pressure and provides auxiliary electrical signals for phase reference. Frequency is continuously adjustable in three overlapping ranges, covering a spread of 72,000 to 1 with upper limit at 20 cycles per second. Amplitude is continuously adjustable 0 to 1 in.; continuously adjustable center position (mean pressure value) is 1 in. Electrical synchronizing pulse is 2 deg at each 90-deg interval of sine wave motion. Sinusoidal electrical output is in exact phase with mechanical motion, and distortion of output motion is negligible at any amplitude or frequency for forces up to 5 lb on reciprocating rod. Made by **Librascope Inc.,** 1607 Flower St., Glendale, Calif.

For more data circle MD-114, Page 177

Portable Oscillograph

Wide-band, quantitative oscillograph, designated type 301-A, measures $9\frac{1}{8}$ x $6\frac{1}{2}$ x $16\frac{5}{8}$ in. deep and weighs 20 lb. Bandwidth extends from 10 cycles per second to 4 megacycles (20 per cent down). Circuits for precision calibration of both time and amplitude are incorporated. Suited to field use, the instrument withstands extreme conditions of temperature and humidity. Primary power requirement is 115 v, 50 to 1000 cycles per second. Made by Instrument Div., **Allen B. Du Mont Laboratories Inc.,** 760 Bloomfield Ave., Clifton, N. J.

For more data circle MD-115, Page 177

Stress Relief

FROM his catalog of engineering specimens, J. P. Henderson presents a familiar type this month.

The Problem Engineer

This is not the type of man to whom the boss gives problems for a quick and useful answer. Instead the man brings his problems to the boss! Sometimes he is known as the complaint engineer, although he is never found behind a window with a "Complaint Department" sign over it.

One variety deals entirely with personal and family troubles. Perhaps it is ill health. Now a successful executive is supposed to have a fairly kind heart and a sincere interest in people. This again is a matter of degree. Given a bedfast employee, I'll be there, telling him honestly not to worry. Or if a man has a wife in the hospital or a sick child, I can be as sympathetic as the next man.

It's all a part of the open-door policy that an effective administrator should maintain. Cold bloodedly, it has been found to pay off in more effective personnel situations and understanding. And if it comes from the heart it's just that much better.

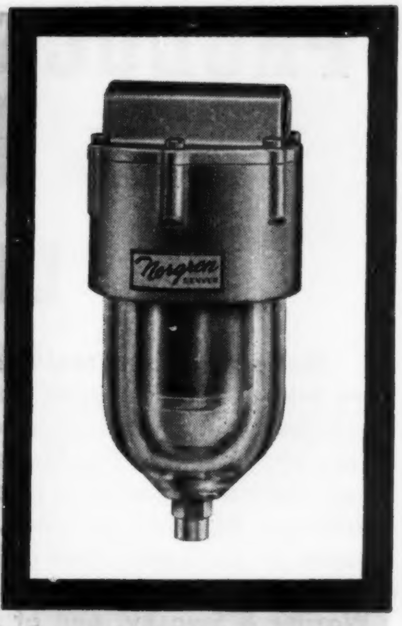
But the regular and detailed problem engineer is a pain in the boss's neck. Weekly or semiweekly accounts of all his personal and family ills can strain any boss's patience.

Another type specializes in the "hurt feelings" problem. I have known engineers who never entered my office to report anything except childish complaints. Someone has hurt his feelings; he has been bypassed or not treated with the respect due him. I make a few clucking noises, which I hope he interprets as sympathy, and wait him out. But what really goes on in his mind, I wonder? Does he think I would really do something because his feelings are hurt?

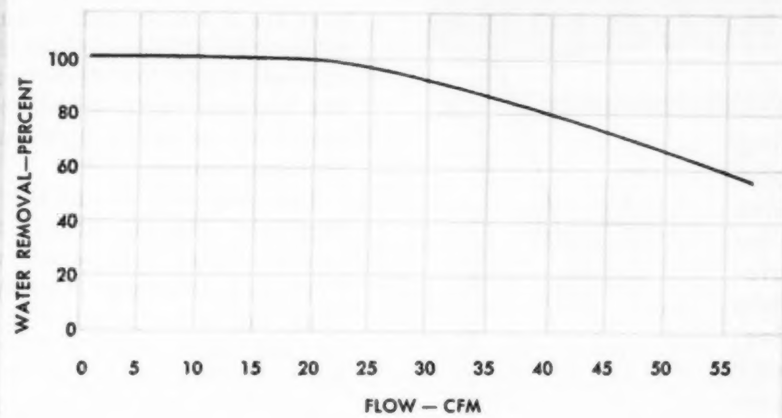
Just what *am* I supposed to do as the boss? The only effective

PERFORMANCE DATA PROVE

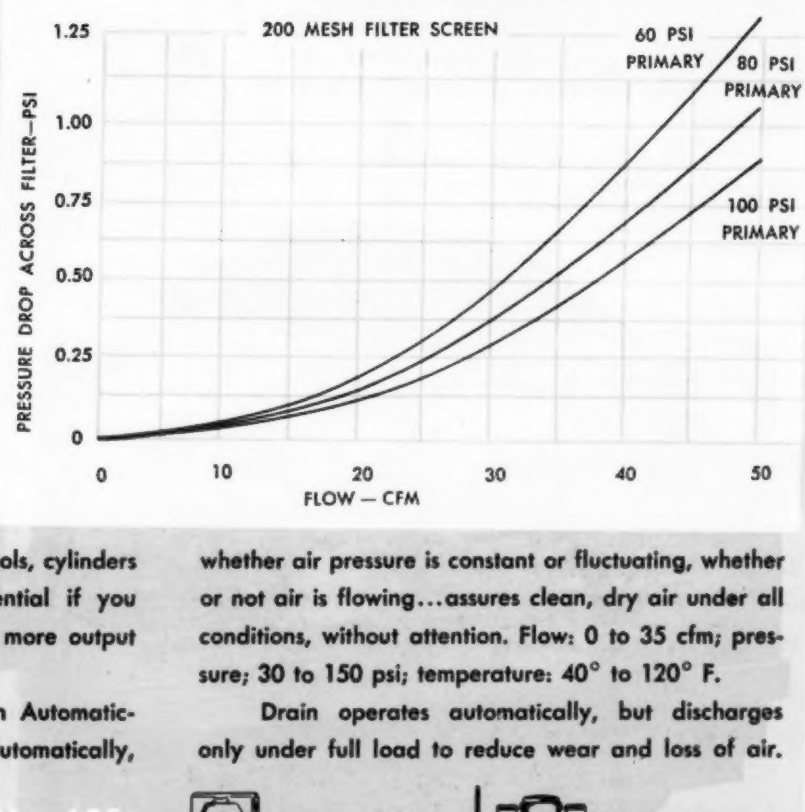
Norgren Automatic-Drain Filter's



HIGH EFFICIENCY IN MOISTURE REMOVAL



LOW PRESSURE DROP



operates whether air pressure is constant or fluctuating, whether or not air is flowing.

Filtering the air that powers your tools, cylinders and other pneumatic equipment is essential if you want better equipment performance and more output with lowest maintenance costs.

The new transparent bowl Norgren Automatic-Drain Filter gives you this vital protection automatically, whether air pressure is constant or fluctuating, whether or not air is flowing...assures clean, dry air under all conditions, without attention. Flow: 0 to 35 cfm; pressure; 30 to 150 psi; temperature: 40° to 120° F.

Drain operates automatically, but discharges only under full load to reduce wear and loss of air.

New transparent replaceable bowl filter for ¾" and 1" air lines. Water capacity 1 pint.

Replaceable metal bowl filter for ¼", ⅜" and ½" air lines.

Permanent metal bowl filter types for ¼" to 1½" air lines.

New, small replaceable bowl filter for ¼" and ⅜" air lines. Screw type clamp ring.

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Stress Relief

action might involve calling the department together and make an announcement something like this: "Now hear this. Stop hurting Joe's feelings. Treat him with respect!"

Wouldn't he be a surprised and hurt Joe if I took such action! It is likely he expects nothing except the opportunity to relieve his mind. The problem engineer merely carries to excess some traits found in all human nature.

The complaint about hurt feelings *is* effective in getting a salary increase. The raise usually comes to the boss, however. In a sympathetic mood, the president is likely to raise the boss's salary for having to deal with such people!

Both the "detail" engineer and the "problem" engineer are great consumers of the boss's time. How much of this time belongs to an employee? This seems to be such an elementary and fundamental thing; any engineer can formulate an equation for it.

The basis for such an equation is very simple if one is within sight of the boss's office. Observe his habits. How much time does he spend per week in regularly scheduled meetings? In informal meetings with other department heads? With outside salesmen? In looking after his mail and dictation? (Judge the attractiveness of his secretary and apply a multiplier to this time.) Add them all up and subtract from 40. How many people report to the boss? Divide them into three classes. The lower group, because of the nature of their work, rate about two minutes per week. The next two groups of semi-important people and section or group leaders rate the rest. With a little calculation one can locate his relative responsibility in the group. How much of the boss's time is really available for a particular individual?

That's right! It's only ten minutes.

The hour-and-a-half week during which the problem engineer has been entertaining the boss with belt slip, abdominal pains, and hurt feelings accounts for the full brief case the boss takes home.

Well, that's one way to get even.

—J. P. HENDERSON

Meetings

AND EXPOSITIONS

Jan. 11-15—
Society of Automotive Engineers. Annual meeting to be held at the Sheraton-Cadillac Hotel and Hotel Statler, Detroit, Mich. Additional information may be obtained from society headquarters, 29 West 39th St., New York 18, N. Y.

Jan. 18-22—
American Institute of Electrical Engineers. Winter general meeting to be held at the Statler Hotel, New York, N. Y. Additional information may be obtained from society headquarters, 33 West 39th St., New York, N. Y.

Jan. 22—
Malleable Founders' Society. General society meeting to be held at Hotel Cleveland, Cleveland, O. Additional information may be obtained from society headquarters, 1800 Union Commerce Bldg., Cleveland, O.

Jan. 25-27—
Plant Maintenance & Engineering Conference to be held at the Hotel Conrad Hilton, Chicago, Ill. Additional information may be obtained from the exposition management, Clapp & Poliak Inc., 341 Madison Ave., New York 17, N. Y.

Jan. 25-29—
Institute of the Aeronautical Sciences. Twenty-second annual meeting to be held at Hotel Astor, New York, N. Y. Additional information may be obtained from society headquarters, 2 East 64th St., New York 21, N. Y.

Jan. 27-29—
Society of Plastics Engineers. Tenth annual technical conference to be held at the Royal York Hotel, Toronto, Ontario, Canada. Addi-

tional information may be obtained from society headquarters, 513 Security Bank Bldg., Athens, O.

Feb. 3-5—

Society of Plastics Industry. Ninth annual reinforced plastics division conference to be held at the Edgewater Beach Hotel, Chicago, Ill. Additional information may be obtained from society headquarters, 67 West 44th St., New York 18, N. Y.

Feb. 15-17—

American Management Association. Personnel conference to be held at the Palmer House, Chicago, Ill. Additional information may be obtained from society headquarters, 330 West 42nd St., New York 18, N. Y.

Mar. 2-4—

Society of Automotive Engineers. National passenger car, body and materials meeting to be held at Hotel Statler, Detroit, Mich. Additional information may be obtained from society headquarters, 29 West 39th St., New York 18, N. Y.

Mar. 15-19—

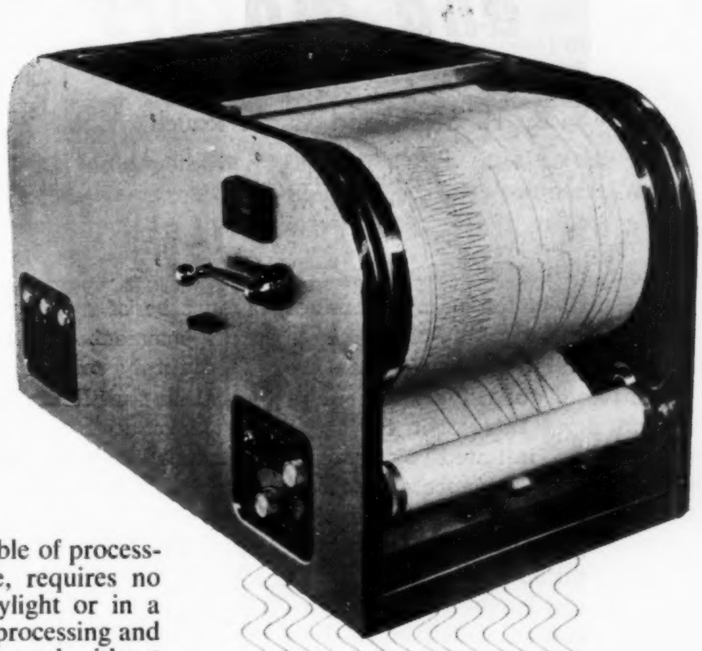
National Association of Corrosion Engineers. Tenth annual conference and exhibition to be held at the Kansas City Municipal Auditorium. Additional information may be obtained from society headquarters, 1061 M & M Bldg., Houston 2, Tex.

Mar. 22-25—

Institute of Radio Engineers. National convention to be held at the Waldorf-Astoria Hotel and the Radio Engineering Show at Kingsbridge Armory in the Bronx. Additional information may be obtained from society headquarters, 1475 Broadway, Times Bldg., New York 36, N. Y.

New, compact photo-record processor

for paper oscillograms and other rolled paper photo-records



THE TEST ENGINEER'S DILEMMA—how to develop and dry large quantities of test data on rolled sensitized paper—is solved by Consolidated's new Type 23-109 Oscillogram Processor. Built for operation at the actual testing site, this self-contained instrument, capable of processing records as rapidly as 15 feet per minute, requires no external water supply and can be used in daylight or in a normally illuminated room. Simplified loading, processing and unloading techniques permit operation by personnel without previous photo-lab experience. The Type 23-109 processor is compact, portable and fast, requiring only electric power for the paper-transport motor and the drying drum. Where the problem is to save time in processing voluminous test data on photo-record paper, CEC's new oscillogram processor is the practical answer. Write for Bulletin CEC 1537A-X5.

Consolidated Type 23-109

- Reduces time lag between test and interpretation of data.
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ANALYTICAL INSTRUMENTS FOR SCIENCE AND INDUSTRY

Design Characteristics of

Ball Piston Pumps and Motors

By H. Stern and W. T. Rauch

Development Engineers
General Electric Co.
Schenectady, N. Y.

THE ball piston pump combines in a compact simple package many desirable features which are indicative of good hydraulic equipment design. Interest in this basic design was initiated by control work on navy ordnance equipment. In this project it was found that servo performance of a gun mount is often limited by the characteristics of the hydraulic pump and motor combination which is used to transmit power from an electric motor to the gun turret. The pump must be of a design in which the displacement can be quickly and accurately controlled to any value within the limits of the design. This requires low mass and short motion of the stroking mechanism in the pump. The motor should have low rotational inertia, so that its acceleration will consume a small fraction of the supply pressure when maximum acceleration of the gun turret is called for. A look at the ball unit design will show how these requirements are fulfilled.

Ball Unit Design: A steel ball is used as a piston. The schematic sketch, *Fig.* 1, shows such a ball in a bore to which oil is admitted at low supercharge pressure through check valve *A* and can be discharged through check *B*. The ball can be made to reciprocate in the bore by a simple cam, rotating about *O*. This scheme completely replaces the piston, crank shaft, wrist pin, connecting rod, and as-

Fig. 1—Basic design of ball piston type pump

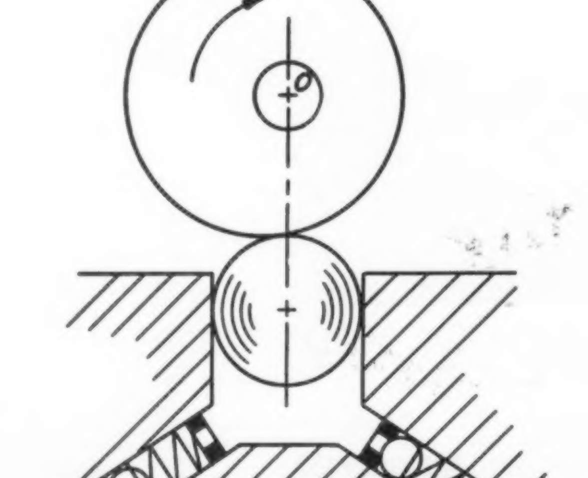
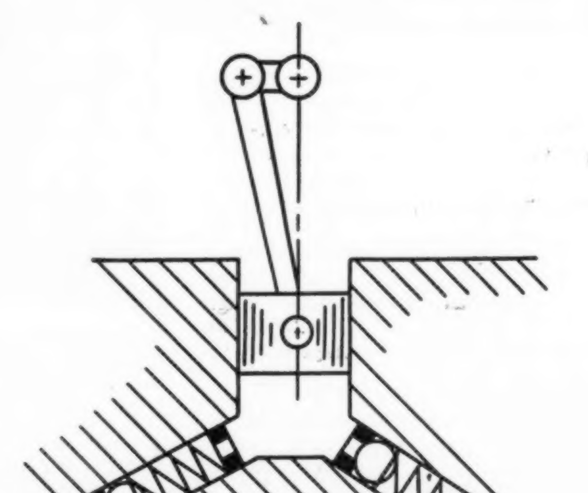


Fig. 2—Basic design of cylindrical piston pump



sociated bearings commonly found in positive displacement pumps, *Fig.* 2.

Multiple Cylinder Unit: Expanding the basic principle of a ball piston pump, a multiple cylinder unit can be constructed as shown in *Fig.* 3. The radial cylinder bore arrangement provides several desirable features:

1. No mechanical piston return is required.
2. Centrifugal force on ball pistons is sufficient to maintain contact between balls and race.
3. Ball-bearing type stroking-ring is easier to machine than a profiled cam.
4. All reaction forces can be supported without use of a thrust bearing.
5. Controlling forces and inertia of moving parts are small.

Referring to *Fig.* 3, the pintle at point 1 serves both as a valve for the oil and as a bearing for the rotating cylinder block, point 2. Oil is taken from inlet passage *A* when a ball travels clockwise around the stroking axis from *X* to *X'*. During this half revolution it moves outward, the space behind it filling with fluid. During the half turn from *X'* to *X*, the ball discharges oil under high pressure into the discharge passage *B*. During the entire revolution the balls, point 5, roll along the eccentric stroking ring, 3, while spinning in their cylinders, 4. Degree of eccentricity of the stroking ring center, *Y*, from the center of rotation, *O*, of the cylinder block determines the excursion of each ball and, hence, the displacement of the unit during each revolution. When the stroking ring center moves from *Y* towards *O*, the displacement decreases until it becomes zero when *Y* and *O* coincide. As *Y* moves past *O* towards *X*, flow is reversed and the function of passages *A* and *B* is interchanged. Maximum stroke for a ball piston is less than ½ the ball diameter, a common value being about 40 per cent. This high bore to stroke ratio of ball piston units allows them to operate at relatively low piston velocities and

Fig. 3 — Design of multiple cylinder ball piston pump

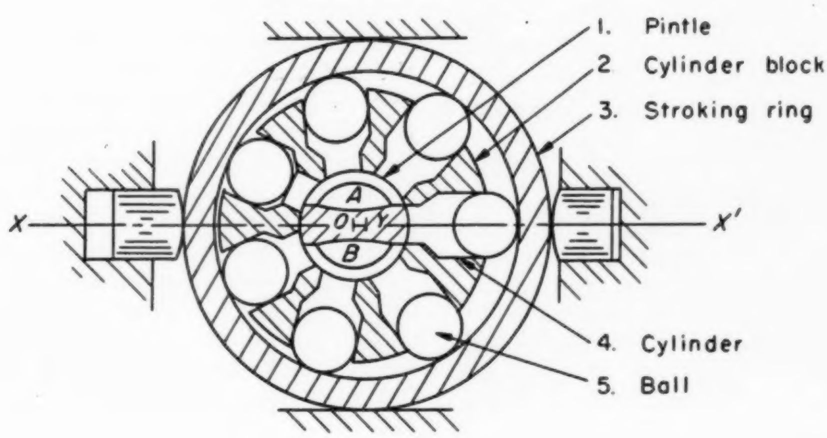


Fig. 4—Ball piston servo pump and electro-hydraulic stroke control produce rapid changes in hydraulic output

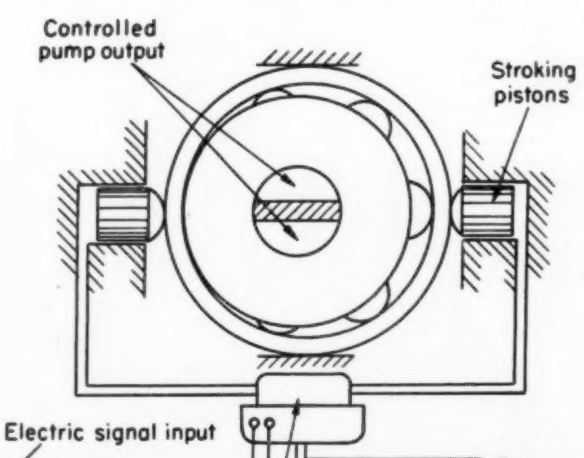
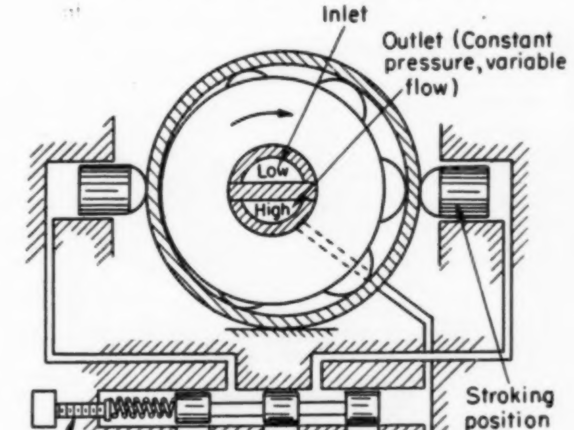


Fig. 5—Cross-section of constant-pressure ball pump which operates at high shaft speeds



avoids high sidewall forces in the cylinder bores.

It need not be stressed that only the highest precision parts made from the toughest materials by intricate manufacturing processes will permit this type of sliding without causing seizures. A ball allows a greater amount of fluid to leak through the piston-cylinder clearance than a cylindrical piston. Proper clearance must be maintained to satisfy both speed and leakage considerations. In ball pumps a small number of high precision parts replace the large number of components commonly encountered in such equipment.

The amount of motion required to change a pump from full stroke in one direction to full stroke in the other direction is small and, therefore, can be accomplished very rapidly by a simple stroke actuator. Small hydraulic pistons of approximately the same diameter as the balls of the unit are sufficient to supply the necessary stroking force. Low inertia of the stroking mechanism permits the unit to vary pressure or flow rapidly.

Two Cycle Design: Radial ball piston units can also be built on a two cycle principle using an elliptical cam in place of the eccentric circular stroking ring. In this case the pintle pin must provide four passage ways for oil, two for inlet oil and two for outlet oil. This design allows each ball to perform two pump cycles per revolution of the cylinder block. Such a pump will deliver twice as much power per revolution as a single cycle unit.

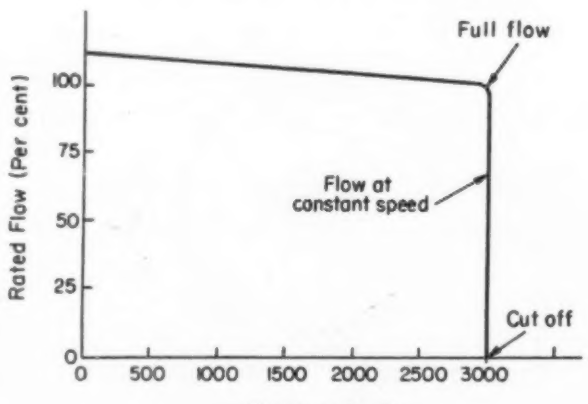
Any ball unit will motor as well as pump. It is evident from *Fig.* 3 that pressure supplied to a ball from passage *A* will cause the ball to move outward in the cylinder bore and thereby supply a clockwise motoring torque to the cylinder block. Ball piston units which are devoted entirely to motoring duty are usually built with the stroking ring fixed at maximum displacement.

Servo Applications: In a ball unit the cylinder block with the ball pistons and the drive shaft form the only rotating members. It is very important that the inertia of these parts can be held low in ball motors. Because of the higher torque to inertia ratio, two-cycle ball piston units are most frequently used as fluid motors. For instance, a typical 0.28-inch^3 per revolution ball motor supplied with 3000 psi fluid will accelerate from standstill to 14,000 rpm in five milliseconds, and this speed is attained in approximately the first ½ revolution of the shaft. This same motor will idle smoothly at low speed with as little as 3 psi pressure drop; initial breakaway pressure is only a few psi higher. These characteristics make the ball motor a useful servo-system component.

The servo pump, which provides the controlled power for such a motor, is shown in *Fig.* 4. In this pump controllability is the major objective, and this is achieved by operating the pump at constant speed and varying the stroke to produce the desired flow. Stroke is generally controlled by an electrohydraulic proportional valve whose input signal level is 3 to 5 watts. A ball piston pump controlled by this device can deliver peak powers of 60 kilowatts; thus power gain of such a servo-pump system is between 1200 and 2000. Low inertia of the stroking ring and the low controlling forces mentioned earlier make it possible to achieve rapid changes in hydraulic output. A ball pump of this type, hydraulically coupled to a fixed stroke ball motor, is now employed to provide the speed gear for a navy gun turret.

Operation at High Pressures: A

Fig. 6—Below—Pressure-flow characteristic of constant-pressure, ball type pump



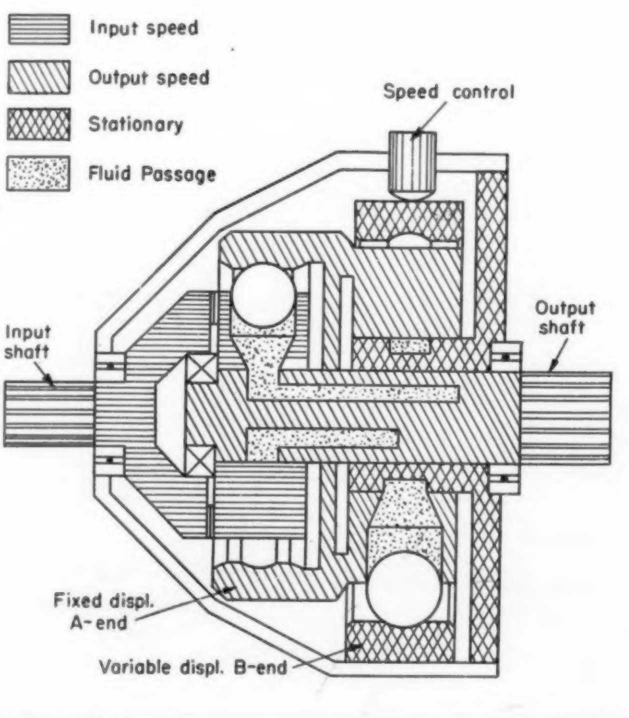


Fig. 7—Right—Differential transmission employing the ball unit type of design. Continuously variable "gear ratios" produced by relative hydraulic displacements of two ball-piston units

constant-pressure pump requires different characteristics than the constant-speed, variable-stroke pump just described. In addition to controllability, efficiency and endurance are required at sustained high pressures for long periods of time. A constant-pressure ball pump must be operated at relatively high shaft-speed in order to displace a large amount of fluid in a given length of time, thereby reducing the quantity of leakage percentage wise. The ball pump not only can achieve operation at high speeds, but it is at these speeds that it excels in performance.

Constant Pressure Design

A ball unit may be operated at ball surface speeds of 100 feet per second. In terms of shaft speed, small units delivering from 2 to 8 gpm are operated at 10,000 to 20,000 rpm. These units may be direct coupled to high speed electric motors or turbines without the use of reduction gears. A constant pressure pump can easily be built with a power to dry weight ratio of 2 hp per lb. Larger pumps delivering 10 gpm and up must be operated at proportionally slower speeds. The largest constant pressure pump presently contemplated will have an operating speed of 7000 rpm delivering 35 gpm at 3000 psi.

The type of so-called constant pressure pump shown in *Fig.* 5 almost delivers a constant pressure as shown by its pressure-flow diagram in *Fig.* 6. The secret of this characteristic lies in the servo control which governs the output pressure. It consists of a small proportional control valve feeding the stroking pistons of the pump. Any change in demand is accompanied by a slight pressure change which in turn gives rise to a movement of the valve. Oil flowing to the pistons changes the output of the pump until original pressure is reestablished. This is an exceptionally fast control system allowing a minimum of overshoot and no steady state error in output pressure. This control permits operation against unprotected shutoff valves without accumulators or

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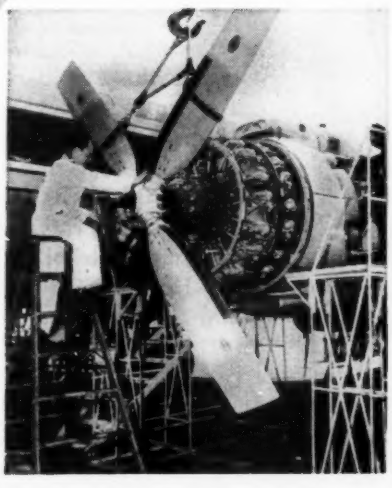

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flexible hoses in the system.

Power to Weight Ratio: Ball piston motors, like pumps, can run at high shaft speeds. They have a place, therefore, in high-speed spindle drives where their good power to weight ratio favors them. For example, small motors weighing about 3 pounds can develop peaks of 30 horsepower at 15,000 rpm. Slip in such a motor is inherently lower than that of a similar ball piston pump, and its efficiency is therefore good even at high speeds.

Transmission Applications: One of the most promising applications of ball piston units is in the field of hydraulic transmissions. Here the shape of the radial piston pump or motor produces a compact package and the complete radial symmetry permits the use of the principle of the differential transmission to best advantage. These transmissions take widely varying input speeds and deliver a constant output speed to drive an aircraft alternator, for example. This is accomplished in a stepless manner, since the transmission ratio is a function of the ratio of hydraulic displacements of the ball piston units involved. By varying one of these, speed can be adjusted continuously.

In a typical differential arrangement possible with the ball unit type of design, *Fig.* 7, the housing of the input unit is locked physically to the cylinder block of the output unit. By varying the displacement of the output end, *B*, the relative speed between *A* and *B* changes. If the *B*-end displacement is large, the *A*-end pumps a large amount of oil through the *B*-end. However, the *A*-end can only pump by having some speed greater than the *B*-end because *A*'s stroking ring rotates at *B*-end speed; therefore, a speed reduction is obtained. If the *B*-end displacement is set at zero, the *A*-end tries to pump but finds its outlet blocked; hence it must run at *B*-end speed. This condition provides straight through drive. Conversely when the *B*-end is stroked negatively, it takes on the function of the pump absorbing mechanical

torque from the *A*-end and delivering high-pressure oil to *A* while it must actually run faster than the *A*-end; thus the unit is in "overdrive." This type of differential transmission is similar to a planetary gear-train with continuously variable gear ratios. The "gear ratios" here are the hydraulic displacements of the two ball-piston units.

Other Design Characteristics: Two forms of such differential transmissions have been built. The larger and earlier design employs two two-cycle units with eighteen pistons each. It is somewhat different from the layout in *Fig.* 8, but operates on the same principle. It is rated at 85 hp output at 6000 rpm and has an input speed varying from 2500 to 8500 rpm. A smaller unit has been built which is rated at 15 hp at 4000 rpm output and will take input speeds between 2000 and 8300 rpm. It uses a two-cycle, ball-piston unit and a single-cycle unit in an arrangement like *Fig.* 8. These differential transmissions are designed for some maximum input torque for all input speeds. Therefore, they are capable of high power-outputs at high input speeds. The ratings refer to the lowest input speed only.

In addition to the advantages of low inertia, high speed, and small number of machined parts required in ball pumps and motors, it was found that these units will operate with fluid possessing a broad range of properties. Units designed to pump 51F21 or MIL-0-5606 fluids have also pumped kerosene without any difference in wear characteristics. Industrial oils have been used in the same pump that ran on silicone fluids—those notorious, low-lubricity compounds. Ball-piston pumps and motors have operated successfully over a temperature range extending from −65 F to 250 F.

These ball piston units have a bright future in high performance hydraulic systems and should be given design consideration.

From a paper entitled "Ball Piston Pumps and Motors" presented at the National Conference on In-

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Electronic Data Processing

By Ira H. Abbott
National Advisory Committee for Aeronautics

WHEN the amount of data obtained in a research laboratory was small, computational difficulties were insignificant, compared with other work. As time passed data processing was handled by centralized computing groups, employing desk calculators of various types. These methods are, however, too inconvenient, too slow and too costly for use in reducing to comprehensible form the great amount of raw data that now flow from our laboratories. As an example of the magnitude of this job, one NACA laboratory measures more than 20,000 pressures on an average day, in addition to a multitude of other types of measurements.

Computation Machines: Coincidently with this growth of data from a trickle to a flood in the last decade, advances have been made in the development of fast electronic digital computing machines with large capacities. The advent of these machines has appeared to promise a solution to the data processing problem by making possible a completely automatic system. One can imagine a system in which the various measuring instruments in a number of wind tunnels and other facilities of a laboratory are connected electrically with a central computing station in such a manner that all measurements would be automatically computed to final form and either tabulated or plotted for analysis and reporting. A question remains, however, as to whether such a system in a research laboratory is desirable.

Large digital computing machines are well suited to analytical work where many mathematical steps are made with a limited amount of input data. Less attention has been given to new tech-

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niques in data handling than to the computational methods with the result that the machines are not as well suited to the task of processing masses of data requiring comparatively few mathematical steps.

Processing Methods: In considering the desirability of a fully automatic system, several factors require attention. The first factor is the high initial cost. It is estimated that such a system for a large laboratory would cost up to several million dollars. Under a realistic system of bookkeeping, the interest on this expenditure should be charged as an offset to the reduction of labor costs. The high cost would also make extensive duplication impractical, so that equipment failure would result in expensive delays especially since qualified maintenance personnel would be difficult to obtain. Major changes in instrumentation would be required throughout the laboratory to assure compatibility between the measuring instruments and the computing machines and this additional factor would be imposed on the already difficult instrument requirements. Flexibility of operation and instrumentation would be sacrificed.

Such a system would require special development for each application, and would make only limited use of commercially obtainable major components. The ability to inspect, edit and control the quality of data at various steps in the procedure would be lost and the completely automatic process might hide or submerge important research information. Finally, unless the system became the master instead of the servant of the research worker, there would always be a non-negligible residue of data to be reduced by manual methods.

Such consideration has led us to the conclusion that it is preferable to approach the problem in a more cautious manner, and to gain experience as we proceed. Much progress has been made in the NACA laboratories during the past year in the mechanization of data processing. Developments

have been along two lines, both of which tend to lead, by divergent routes, to more fully automatic systems.

First, semiautomatic methods of processing data have been introduced that are flexible in the sense that all kinds of data can be handled regardless of the source, type of data, or method of measurement.

Secondly, fully or nearly fully, automatic methods have been developed for processing or partly processing specific types of measurements that are made in large numbers in a standardized manner.

Semiautomatic Processing: The Langley Laboratory is now processing data at the rate of 3,000,000 points per year and is employing the semiautomatic method. This data processing section is gradually replacing manual computing, and is capable of handling data from all activities, such as wind tunnels, flight research on airplanes, and flight research on rocket-propelled models. The raw data, together with instructions for processing, come to the group in various forms including manually prepared data sheets, printed tapes, and traces on film from various types of instruments. In normal operation, the data are completely processed and returned to the originating research unit within 24 hours. In special cases, arrangements are made for more rapid processing. For instance, a messenger may deliver a batch of raw data every 2 hours, and pick up the processed data delivered on the previous trip.

Regardless of the form in which the data are received, all information is transferred to punched cards. In the case of data sheets, etc., this operation is performed by an operator using a standard cardpunching machine. If automatically punched cards are received, any additional information needed for the processing is added by the card-punch machine. When the information is received on film, extensive use is made of film-reading equipment that prepares the cards directly from the photo-

Design Abstracts

graphic records. With this equipment one person can prepare from 3000 to 6000 cards per day. Cards are then processed in calculating machines, and the data are tabulated in forms desired.

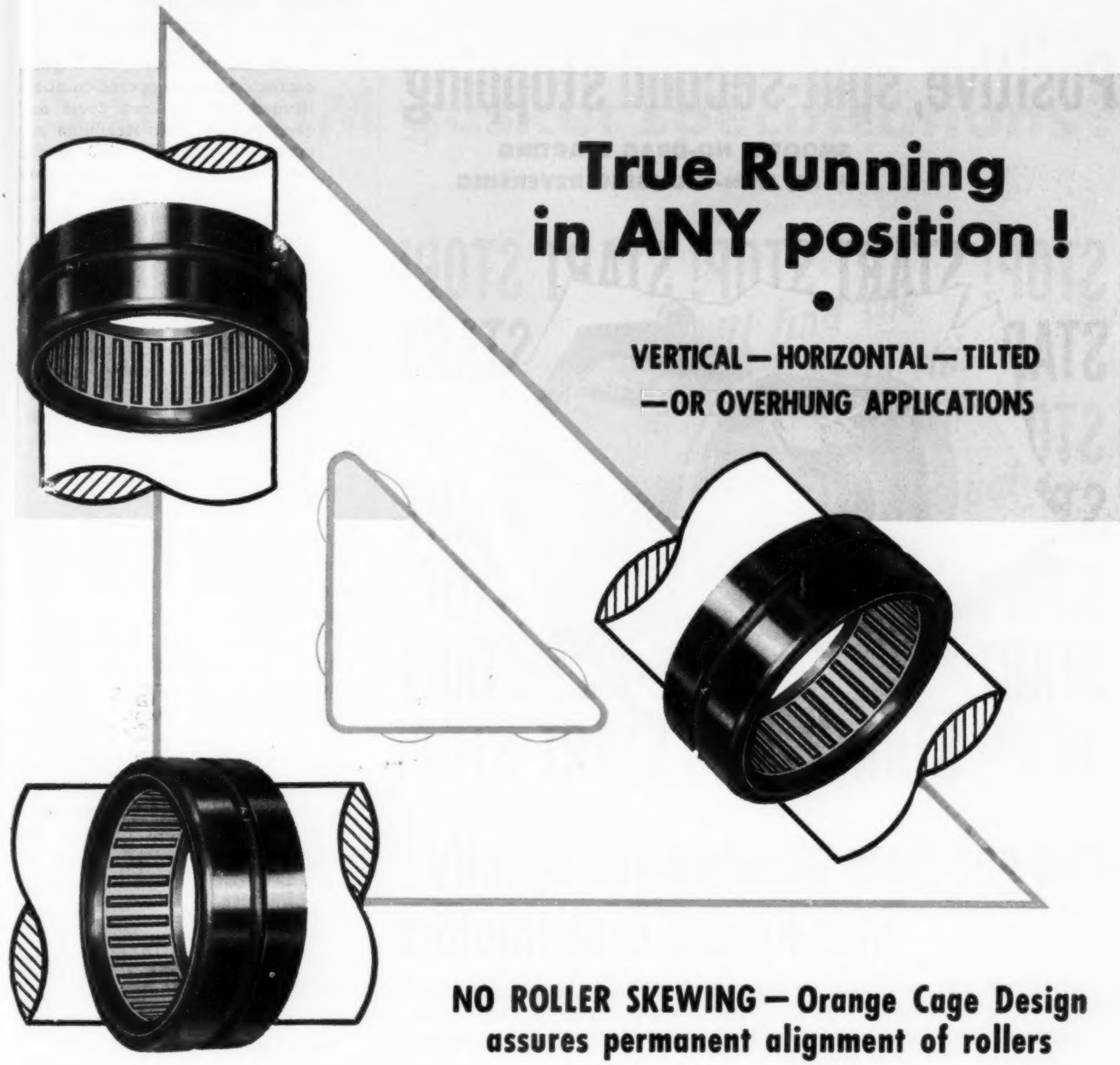
An increasing proportion of the data obtained is processed in this manner, and it is believed that this system is more economical than either manual or currently available high capacity computing machines. Personnel requirements and costs of the semiautomatic method for the past year were compared with manual methods using desk calculators. It was noted that personnel requirements were less than one-fourth of that required for manual operation, and cost was slightly more than one-half. Cost figures included machine rentals and depreciation of owned equipment. These figures were averages for the first year of operation, and some recent months have shown substantially larger savings.

Advantages of the semiautomatic system may be summarized as follows:

1. Substantial manpower savings are realized.
2. Data are available for analysis sooner than by manual methods resulting in more efficient use of research facilities and personnel.
3. System maintains the ability to inspect, edit, and control quality of data at various steps in process.
4. System is economical.
5. Equipment is commercially available at reasonable cost.
6. System is flexible and is easily adapted to various jobs and to continually changing instrumentation and testing techniques.

Automatic Processing: More fully automatic equipment was developed for a specific purpose at Lewis Laboratory. A digital automatic, multiple-point, pressure-recording system was installed to process several million pressure readings taken there each year. Some of the basic considerations in the development of the system were the following:

1. High accuracy and wide range were necessary for measure-



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ments of pressures on engines corresponding to operation at altitudes between sea level and over 50,000 feet. Readings accurate to within 1/2000 of the full range of the instrument were desired. It was felt that this accuracy could be obtained most easily through digital methods.

2. Equipment associated with each individual pressure had to be simple and inexpensive; more complicated measuring equipment should serve 100 or more pressures.
3. Equipment should feed into commercially available punched-card calculators that could perform long and complicated computational routines.

In this system, pressures are measured by comparing them to an accurately measured variable reference pressure.

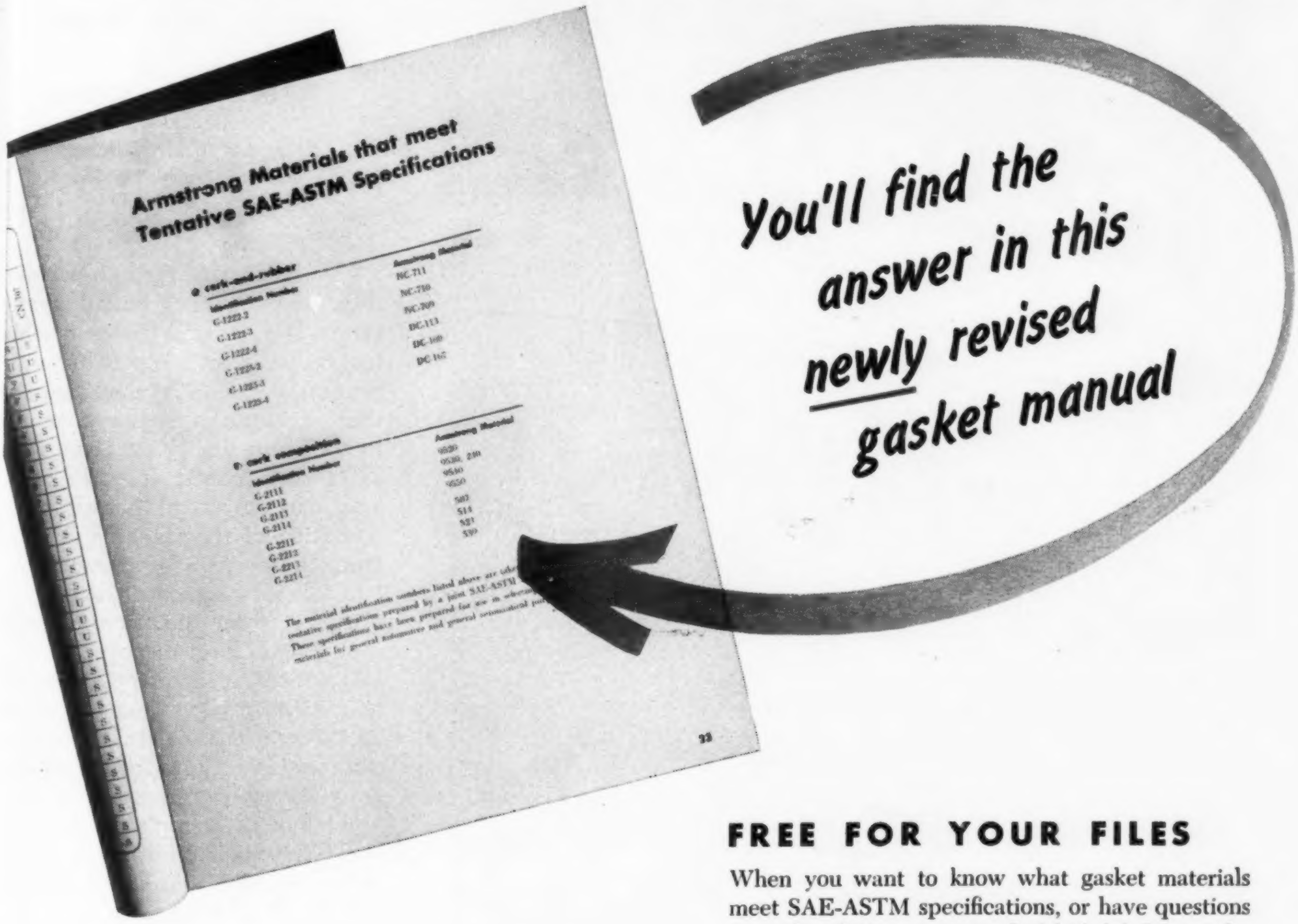
The machine was completely automatic in operation with alarms and signals in case of malfunctions. Symbols were punched into tape if a reading was to be disregarded because of instrument trouble. A complete cycle took about 2 ½ minutes; however, it was only necessary to hold test conditions constant during 10 seconds of pressure-scanning. After this, the test engineers could proceed to establish the next test condition as the read-out operation proceeded.

Data Plotting

To complete the data reduction process, it is necessary to plot the computed result in various ways for analysis. Prior to the use of semiautomatic methods for reducing data, it is estimated that one-fourth to one-third of the processing time was spent in plotting results. Now that machine methods are used, plotting requires an even greater portion of the total time. It is apparent that machine methods of curve plotting would be a great aid in further reducing the delay between test and analysis of results.

From Advisory Group for Aeronautical Research and Development Memorandum, "Methods Used by NACA for Data Reduction", presented at Rome AGARD Con-

What Armstrong materials meet SAE-ASTM gasket specifications?



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ference, December 1952.

Tracer Controls for Machine Tools

By George L. Rogers
Specialty Control Dept.
and John L. Dutcher
Industrial Engineering Sec.
General Electric Co.
Schenectady, N. Y.

AN ACCURATE, high-performance, tracer control system cannot be treated as a machine tool attachment. For satisfactory results, it must be designed into the machine, with numerous features being added or changed to allow the tracer to perform at its best and to make the setup and operation of the machine practical from the operator's viewpoint.

A tracer-controlled machine is different from the usual machine tool in two basic respects. First, it is operated in a different manner and therefore various operating features are required which are not needed on standard machines. Provisions for efficient set-up of tracing head, template, and work must be made so that the time saved in machining is not lost during set-up operations.

Secondly, a different concept of mechanical design is required. Machine deficiencies such as vibration, stickiness, and backlash produce different effects when a machine is tracer-controlled than they do when it is under the control of an operator. While the tracer can compensate for these deficiencies under many conditions, it may tend to exaggerate their effects under others. Since the tracer-controlled machine is asked to cut all shapes including many that would not otherwise be attempted, the correct set of operating conditions to make any of these faults evident will be attained sooner or later.

Types of Tracer Systems: There are a number of basic types of tracer systems. Single-dimension tracers normally used on lathes are the most common. Two-dimension tracers are used on milling

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machines for profiling work and also on vertical boring mills. Selective two-dimension tracers, which are employed for die sinking or similar machining operations, provide tracing in any one of the three principal planes of the machine with the selection of the operating plane being made by simple switching methods. Simultaneous three-dimension tracing may be accomplished by combining a two-dimension and a single-dimension tracer and using either two tracing heads and two templates or a single combined tracing head with one three-dimensional template.

All tracers whether they be hydraulic, pneumatic, mechanical or electric, are made up of essentially the same parts in one form or another. Each tracer includes a sensing element which feels the surface of the template and provides a form of intelligence to the tracer. It includes amplifying elements which amplify and generally modify this intelligence to make it useful in controlling drive elements which power the machine feeds. Also a real part of the tracer system is the mechanical gearing or linkages which transmit power from the drive elements to the linear motion of the machine feeds. Generally one form or another of stabilizing means will be used to damp out oscillations or overcorrections which may tend to creep into the system.

Tracer Deficiencies

Many deficiencies exist in tracer systems and machine tools which introduce errors. These may be broken down into three types:

1. Resolution errors due to small imperfections in the electrical or mechanical system for which the tracer must compensate. This compensation requires a change in stylus deflection and consequently results in an error.
2. Errors on corners or sharp curves due to the inability of the feed motor to change speed instantly.
3. Errors caused by machine imperfections which are outside the tracer control loop and therefore cannot be compensated for or which upset the tracer system

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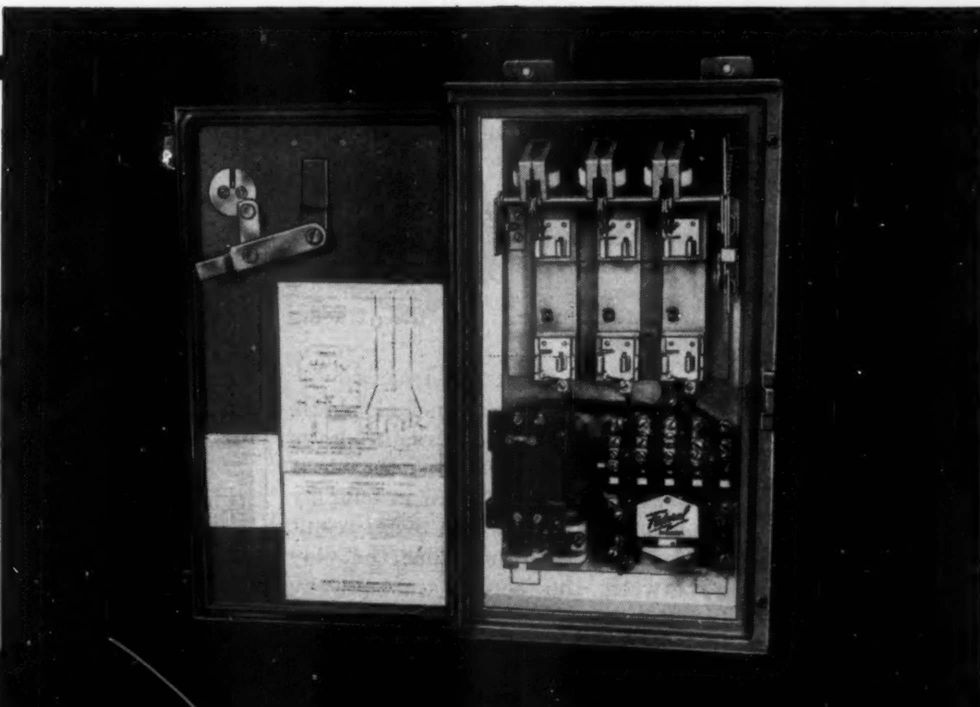

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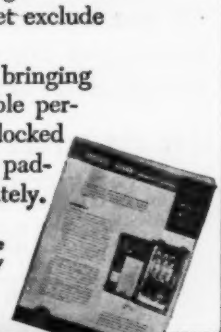

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and cause transient errors.

Everything possible is done to the electric system to compensate for both its own deficiencies and those of the machine. However, since compensation can never be complete, it is essential that machine deficiencies be eliminated or reduced to the absolute minimum. These deficiencies are:

1. Backlash in the feed drive
2. Windup in the various parts of the feed drive system
3. Stickiness of the machine ways
4. Vibration
5. Deflection of the machine members
6. Misalignment of the tracing head.

Backlash: Two principal places where backlash occurs in large amounts are between the lead screw and the nut, and in endplay of the leadscrew. The usual result of an attempt to reduce the play between the screw and nut by tightening double, split, or other adjustable nuts is to bind up the drive at some point on the screw while it runs freely at another. This condition can be improved by increased precision in the machining of the screw and nut. The ultimate may be the use of a ball bearing nut which can be preloaded heavily without binding.

Backlash introduced by endplay in the leadscrew or nut, whichever is the rotating element, can be reduced to a small value by good preloaded bearing techniques. Ball and roller thrust bearings can be adjusted to essentially zero endplay.

Windup: Elastic deformation of machine elements caused by the forces required to drive the machine motions produces windings. It may stem from twist of screws and shafts or deflection of their supports. It is characterized by a spring effect and may be minimized by the use of closely-coupled, sturdy parts, and by reducing friction opposing movements of the machine members.

Stickiness: Variation in the friction between sliding surfaces causes stickiness. It results most-

ly from the difference between the force required to initiate motion and that required to maintain motion. It is affected by the nature of the sliding surfaces and the lubricant used. Good, smooth bearing surfaces lubricated with special high-pressure oils help tremendously. For any given machine design, considerable study may be required before stickiness is eliminated.

Vibration: In machine tools using tracer control, vibration breaks down into two problems—its reduction and prevention of its transmittal to the tracer head and template. Improvement in geared drives and increased rigidity of machine parts will help reduce vibration. Rigid mountings for the tracer head and template will help eliminate vibration amplification by these parts.

Deflections: Machine deflections can be reduced by increased rigidity. Their effects can be decreased by good tool-tracer, work-template arrangements. For an example of the latter, if the arrangement of the tool-tracer combinations is such that a deflection moves the tool in one direction and the tracer in the opposite direction (not usual), the deflection effect is multiplied by a sizeable factor.

Alignment: The aspect of machine alignment is not generally a serious problem because it is important in the design of the machine tool whether a tracer is employed or not.

Tracer Control Performance: Machine tools with tracer control have performance characteristics closely related to the machine condition and the feed speeds involved. At higher speed the adverse effects of the deficiencies in control and machine are much more pronounced. However, tolerances and finish requirements are also generally relaxed. At lower feed speeds where requirements are usually most stringent, operation improves until at extremely low feed rates the effects of inherent machine and control deficiencies

essentially disappear.

HIGH SPEED MILLING: At the upper extreme of the range of tracing feed speeds which have been observed, a small milling machine is capable of machining at feed rates up to 100 inches per minute. This machine has a backlash of 0.007-inch, little windup, and practically no stickiness. At 100 inches per minute the machine will follow shapes having curvatures of one inch radius with an accuracy of plus or minus 0.015-inch. The limiting factor under these conditions is the machine backlash.

At lower feed speeds, performance on the same machine improves until at 0.5-inch per minute the accuracy is plus or minus 0.015-inch. The limiting factor under these conditions is the machine backlash.

At lower feed speeds, performance on the same machine improves until at 0.5-inch per minute the accuracy is plus or minus 0.005-inch over an almost unlimited series of template shapes. Backlash again is the limiting factor.

LOW SPEED TURNING: At the extreme low end of the range of feed speeds are the large boring mills used for difficult machining of materials such as stainless steel. The performance of several of these machines has been observed on production work.

Surface Finishes

One such machine, which was employed to turn a ball surface approximately 36 inches in diameter with an interrupted cut, was operated at a feed speed of 0.125-inch per minute. On a cut of about 8 inches which required more than an hour to complete, the finished surface was essentially polished with a slight mark where one motion reversed through the 0.015-inch backlash of the machine. This mark, as nearly as it could be measured was 0.0005-inch and consisted entirely of excess material.

Another series of machines turning jet engine rotors have consistently been producing finishes of 50 microinches RMS on gentle compound curves. Small backlash marks are evident but not ob-

jectionable.

One automatic machine operation successfully runs through a sequence involving feed speeds of about 2 inches per minute while cutting, with a speedup to 75 inches per minute for traverse between cuts. No cutting was done during this traverse operation, but the tracer was required to maintain contact with the template and be ready to cut immediately at the end of the traverse operation. Accuracies better than 0.0005-inch were maintained.

From a paper entitled "Practical Considerations in the Use of Tracer Controls" presented at the Sixth Annual AIEE Conference on Machine Tools in Cleveland, O., October, 1953.

Metallic Traction in Transmission Design

By L. A. Graham
Mechanical Engineer
Graham Transmissions Inc.
Menomonee Falls, Wis.

THE most common use of metallic traction to transmit power is in the railway locomotive. Early designers of railway equipment evidently feared that traction would not push the train ahead because there are interesting pictures in the texts of a century or so ago of a third, toothed rail or rack alongside the other two rails, which was engaged by a gear driven by the steam engine. However, some bold spirit soon came along and threw the rack into the discard, and traction has been doing the job ever since.

Some of the early automobiles were traction driven to an intermediate dead axle by a roller splined to the engine shaft and engaging under spring pressure a disk, usually with leather facing. The roller was moved along the disk by a linkage to give all speeds, forward and reverse. Final drive to the rear axle was by chain. When the going got too rough, or too tough a grade was attempted, the engine had plenty of power to keep turning the roller but there was not enough traction

for it to turn the disk, so the roller would merely grind a hole in the disk and the car just wouldn't "make the grade".

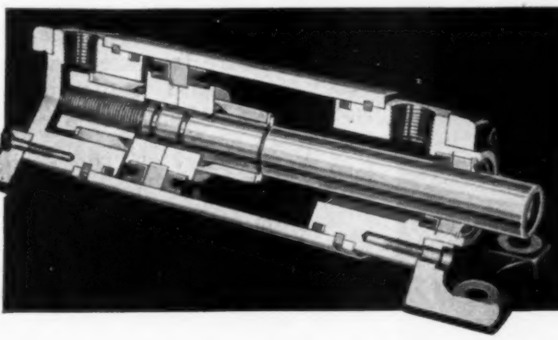
There are many such elementary roller-disk drives still in use on certain types of industrial machines where the service demands are not too rigid, but this construction violates most of the numerous requirements of a well-designed traction drive. Largely because of this poor initial record, even well-designed modern traction drives are still held in disfavor with many engineers. They are often dubbed "friction" drives, the word "friction" being used as a term of opprobrium, which Webster defines as "reproach mingled with disdain." In fact, one such machine-tool designer remarked that he wouldn't have a friction drive in his machine. It happens, however, that the machine-tool spindle has always been driven from the motor, and with good success, by V-belts whose design follows the familiar formula $T_2/T_1 = e^{f\theta}$, f being the coefficient of friction or traction which performs the job. Actually, traction properly applied participates in driving most modern machines, and metallic traction is widely called into play when the requirement is for infinitely variable speed in which it does an accurate and reliable job.

Dry vs. Lubricated Traction: When alloy steel traction members in contact under pressure are used instead of a belt and pulley there is the basic advantage that alloy steel has over thirty times the strength of leather. This means that parts can be smaller and the transmission more compact even with a smaller friction coefficient. Since steel parts can run in oil, the transmission can be self-contained and splash-lubricated. Dry metallic traction, although it naturally affords a higher operating coefficient, can be used for power transmission only in the locomotive where the rails extend for indefinite length, or on small intermittent operation "fleapower" units. The relatively low friction coefficient resulting from the im-

mersion of the traction members in a lubricant calls for the use of rather high contact pressures comparable to those in ball bearings. This leads to the most important single requirement of any successful, commercial variable speed transmission of this kind, namely: the construction must be such that a complete skid at the contacting parts is impossible even in the event of a complete stall from overload of the driven shaft such as took place in the early disk-roller automobile transmissions. A complete skid at the contact under the existing and allowable pressure would mean an almost instant destruction of the surfaces. This can be prevented only by building a gear differential into the inherent design.

Metallic Traction Laws: One of the interesting features of lubricated metallic traction is the uniformity of the laws that govern its use. In lubricated traction the action is quite different from dry traction. In dry traction there is a microscopic interlock of the engaging surfaces developed by the pressure; this interlock holds the surfaces firmly together and breaks down only when the friction coefficient is exceeded. With lubricated traction, a monomolecular oil film is always present between the surfaces, and its stretch transmits power up to the limit of traction. This stretch of the oil film results in a slight decrement of speed corresponding to each increment of traction load, the curve being comparable in shape and characteristic with the speed-torque curve of an induction motor.

Since both tractive power and regulation are tied up in the oil, the oil used in a traction drive should be selected for traction rather than lubricity, although adequate lubricity must be provided. Slight loss of speed under load or slip, which is characteristic of the induction motor and makes it work, is also characteristic of any infinitely variable-speed transmission not using an auxiliary governor whether it be a mechanical, electrical, hydraulic



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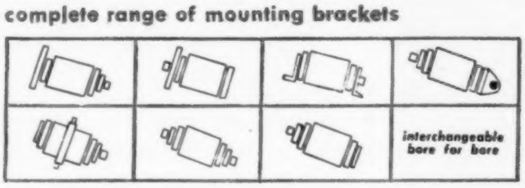

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Design Considerations: In the use of metallic traction for modern transmissions, a paramount design requirement is to determine mathematically the stresses at the frictional contact and the resulting life expectancy. These formulas were worked out originally by Herz in 1893 and are the foundation of all modern bearing and gear design. They enable the designer to calculate the dimensions of the flattened elliptical area of contact and thus obtain a measure of the surface shear. An empiric factor, however, which must modify such calculation is that of temperature. It is important to keep the case temperatures comparable to those in electric motors —not more than 50 C above ambient. Fatigue tests made on bearings in refrigerated cases have shown that failures are basically a heat phenomenon, thus reduced temperature increases fatigue life.

Summary

A summary of the design requirements in applying metallic traction to a modern transmission are the following:

1. Make it physically impossible to completely skid.
2. Use proper tractional lubricant.
3. Design for highest regulation.
4. Keep operating temperatures within bounds.

The last two requirements are met by the following features of transmission design:

1. Limit contact pressure to moderate values.
2. Locate tractional surfaces at extreme case dimension where leverage of traction is greatest.
3. Construct drive for minimum side spin to permit least possible variation of speed at sides of contact center.
4. Minimize number of tractional contacts to insure load division and to lower cost per horsepower output.

From a talk entitled "Use of Metallic Traction in Modern Transmissions" given before the Machine Design Division of the Southern California Section of ASME in Los Angeles, Nov. 1953.

New Machines

Materials Handling

Hydraulic Crane: Rated at 3200 lb capacity at a boom radius of 18 ft and 8000 lb at a 10-ft radius, unit can lift and transport its maximum load any distance indoors or outdoors. With boom extended, hook can be raised 24 ft or lowered 30 ft below ground level. Telescopic boom can be raised to any point between the horizontal and 45 degrees and can be turned continuously through 360 degrees. Employs standard 6-in. center distance, 50:1 Cone-Drive gearset in the boom lifting mechanism; 5-in. center distance, 40:1 ratio gearset on the crane swing drive. *Austin-Western Co., Aurora, Ill.*

Electric Fork Truck: Has extra short wheelbase, 43½ in., to permit turning in crowded areas; 68-in. high uprights and full initial lift to permit stacking in low head room areas; and heavy-duty drive axle to withstand high speeds and fast acceleration. Contactor control permits fast, smooth starts with one handle controlling four speeds in both directions. Hydraulic lift mechanism features controlled lowering and an automatic unloading device to prevent overloading. *Elwell-Parker Electric Co., Cleveland, O.*

Vacuum Lift: Handles nonporous metal sheets up to 40 x 40 in. at the rate of one to six sheets per minute. Picks up at any level from floor to 36 in. high; is adjustable through this range and through 180 degrees turn. Per cycle air consumption is approximately 60 cu in. Machine contains its own vacuum plant. Floor space required is 36 x 48 in. for base, with arm extending out approximately 4 ft; height is 6 ft; weight, 1800 lb. *Union Tool Corp., Warsaw, Ind.*

Lift Trucks: Model SD, with capacity of 2000 lb, handles either single or double-faced pallets. Length, excluding forks, is 13 in.

Lowered height of truck is 3¼ in., full lift is 4 in. Overall width is 22 in.; width of each fork, 7¼ in.; opening between forks, 7½ in. Model D, also with 2000-lb capacity, is used with double-faced pallets. Overall length of truck, excluding forks, varies from 13 in. to 24 in., depending on length of forks. Overall width and fork dimensions same as for Model SD. Forks are designed to enter a vertical space of 3⅝ in. Forks for both models are made from 24 to 72 in. long. Both trucks require less than a 30-lb pull on a level hard concrete floor and less than 25-lb effort on the lifting handle to raise a load. *Market Forge Co., Materials Handling Div., Everett, Mass.*

Metalworking

Straightener: For heavy-duty use, model 3BC straightens steel and nonferrous tubes and round bars such as heavy walled seamless tubing or alloy heat-treated hot-rolled bars. Straightens tubes from 1 to 10 in. OD and bars from 1 to 6½ in. diameter at speeds up to 320 fpm. Any 4:1 ratio can be furnished for particular applications. Machine employs a central pressure roll located between two sets of opposed rolls, each set having one driven roll and one idler roll. Angular adjustment of the two driven rolls is shown on two dials. Size: 20 ft long, 15 ft wide, 10 ft high, including drive. *Sutton Engineering Co., Bellefonte, Pa.*

General Purpose Machine: Double end machine, does boring, chamfering, centering, burring, milling, flaring and spinning operations. Hydraulically powered magazine feed and motorized heads equipped with hydraulic feeds make possible production of more than 6000 pieces per hour. Spindles in heads resist unbalanced thrust loads up to 14,000 lb. Workpieces are loaded in magazine, fed one at a time and chucked in a stationary position for machining by action of a hydraulic cylinder which controls fingers and chuck jaws at each end of the workholding fixture. Two air cylinders act in conjunction with the hydraulic cylinder to pro-

MACHINE DESIGN

18th Edition

Directory of Materials

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The "Directory of Materials" will be published as a supplement to MACHINE DESIGN's Special Materials Issue in May of this year. The editors have conceived, planned and organized the Directory so that it will be possible to "get the facts" by having only a fragment of information to begin with, whether it be the type, tradename or the producing company.

TYPE: The Directory will contain complete listings of materials in these groups: Metals, Nonferrous Metals, Nonmetallic Materials and Plastics, and Finishes and Coatings. All the major alloys and related varieties will be listed under each type.

TRADENAME: Within each section there will be an INDEX OF TRADENAMES, containing detailed information on service characteristics, producibility characteristics and specification designations.

COMPANY: Following these sections will be an "index of the indexes" . . . an alphabetical listing of all tradenames, a listing of all participating companies and a listing of specification agencies. Each tradename and each company will be cross-indexed to the exact place in the Directory where all the other pertinent information is to be found.

vide uniform tension throughout loading and unloading. Power heads rest on flat ways on machine bed and are adjusted to a position to suit length of part. Cutter spindles automatically compensate for part length variations and are hydraulically fed into depth and retracted. Finished part is ejected as a new part drops into position. Adjustable stops control travel of cutter spindles, which is adjustable up to 6 in. *Walter P. Hill Inc., Detroit, Mich.*

Automatic Screw Machines: Two $4\frac{1}{4}$-in. capacity machines differ in type of feed drive. Model AB has electric feed drive which provides adjustable forward and return motions of each of the five turret stations. Drive is controlled by ten dial rheostats and a rotary timing switch with ten selector contacts. Model AW provides variable forward and return feeds through a mechanical drive. Changes of feed are automatically controlled by adjustable steel cams on the regulating wheel. Ranging from 21 to 648 rpm, 56 spindle speeds are available. Four automatic speed changes for each speed range are made possible through the four-speed spindle motor. Hand shift clutch provides two speed ranges for each set of change gears. Separate front and rear cross slides are independently operated by separate, adjustable cam drums. Tool turret diameter is 14 in.; stock feed stroke, 14 in.; turning length, $7\frac{7}{8}$-in. *Cleveland Automatic Machine Co., Cincinnati, O.*

Impact Press: All-pneumatic press features uniformity of impact force on workpieces of varying thickness and speeds up to 10,000 cycles per hour. Variable-stroke air hammer or piston ram which moves within a closed cylinder strokes a chuck projecting through the bottom of the cylinder. Length of hammer stroke or travel inside cylinder is adjustable. Actuation of foot-valve pedal brings cylinder assembly with chuck and tool into contact with work by a hydraulic traverse mechanism, and impact follows. Traverse and impact stroke are in linked sequence, but independently adjustable. On work of uniform thickness clear-

ance of 0.005-in. will permit an impact of any desired force ranging from 30 to 30,000 lb. On work that varies considerably in thickness, vertical clearance may be set for the thickest piece, and within the traverse range of 2¼ in., impact is uniform. Cylinder bore is 3 in.; piston travel or hammer stroke, 0 to 4 in.; throat clearance, 8½ in.; vertical clearance, 0 to 12 in. *Heidrich-Nourse Co., Los Angeles, Calif.*

Gear Shaver: Improved horizontal model GCK-96 for spur, helical and herringbone gears from 2 to 16 diametral pitch, having pitch diameters from 24 to 96 in. Has two cutter heads with rapid traverse feeds and a flexible work driver, and adjustable bearing pedestals which permit shaving of large gears by the rotary crossed axes principle while mounted on their journal bearings as they will run when installed. Bed supports gear assemblies weighing up to 26,000 lb. Checking bar mounted in a slide in the saddle which supports the two cutter heads checks alignment of gear journal bearings in both horizontal and vertical planes. Each cutter head has indicators for setting the cutter head crossed axis angle. Accommodates cutters up to 12 in. in diameter. Maximum shaving range is 111 in. Length from headstock to end of work bed is 170 in. Machine is 240 in. long, 210 in. deep and 129 in. high, and weighs approximately 150,000 lb. *National Broach & Machine Co., Detroit, Mich.*

Processing

Automatic Electroplater: Incorporates only longitudinal and vertical reciprocating movements. Has neither elevating superstructure, transfer cams, chain, sprockets, nor hydraulic fittings above the tanks. All working parts are mounted on a central carriage which moves on a single track and is actuated by a hydraulic cylinder. A shaft running the length of the carriage is fitted with gears meshed to racks on the work lifting arms, providing vertical motion which lifts and lowers work. As lifting arm racks are meshed with central shaft gears on opposite sides, arms on one side

New Machines

are raised while those on the opposite side are being lowered. Work carriers which are independent of the transfer mechanism move on two cathode rails. Before unloading, parts are moved through a drying oven. *Wagner Bros. Inc., Detroit, Mich.*

Vacuum Forming Machine: Model 50-20, for vacuum forming thermoplastic sheets, operates with completely automatic cycle and has 90-degree adjustable platen for differential heating of the sheet. Especially suitable for the use of male dies that permit deep draws with relatively thin sheets. Machine has 60 x 30-in. platen which handles sheets up to 52 x 24 in.; height-adjustable strip heater assembly that provides temperatures up to 1150 F; 100-gal vacuum surge tank; and cycle control by means of four delay timers. Overall dimensions are 88 in. high, 70 in. wide, 68 in. deep; weight is approximately 2500 lb. *Vacuum Forming Corp., Port Washington, N. Y.*

Parts Cleaner: Model H-95, for flushing, spraying or soaking metal parts, dies, large bearings and assemblies. Pump delivers flow of cold solvent under pressure through a heavy flexible metal hose. Selector valve permits direction of flow by operator also. Pump can circulate fluid in the tank by jet action. Solvent capacity is 100 to 175 gal. *Graymills Corp., Chicago, Ill.*

Vacuum Metallizer: Rapid pumpdown of 72-in. diameter, 60-in. long chamber is accomplished by a three-stage pumping system, comprised of a 500 cfm rotary mechanical pump for roughing, a 6-in. booster pump and three 14-in. oil diffusion pumps which are directly connected to the rear face of the metallizing chamber, without a manifold. Pumping system can evacuate the empty chamber to a pressure of ½-micron—normal working pressure for metallizing — in 7 minutes. Pushing a button to initiate rough pumping automatically closes all valves opening from vacuum chamber except the one to mechanical vacuum pump. When rough vacuum pressure gage reading is 200 microns, operator pushes a second


Foot mounting . . air
Pendulum mounting . . air
Rabbet mounting . . air
Foot mounting . . hydraulic
Pendulum mounting . . hydraulic
Rabbet mounting . . hydraulic
Base mounting . . air
Trunnion mounting . . air
Flange mounting . . air
Base mounting . . hydraulic
Trunnion mounting . . hydraulic
Flange mounting . . hydraulic


Centerline mounting . . hydraulic
Double end rod mounting . . air
Double end rod mounting . . hydraulic

button to close valve from chamber to roughing line and open valves to diffusion pumps and booster pump. After metallizing takes place, depressing a third pushbutton closes all valves leading out of the chamber and trips open another valve which admits air. *F. J. Stokes Machine Co., Philadelphia, Pa.*

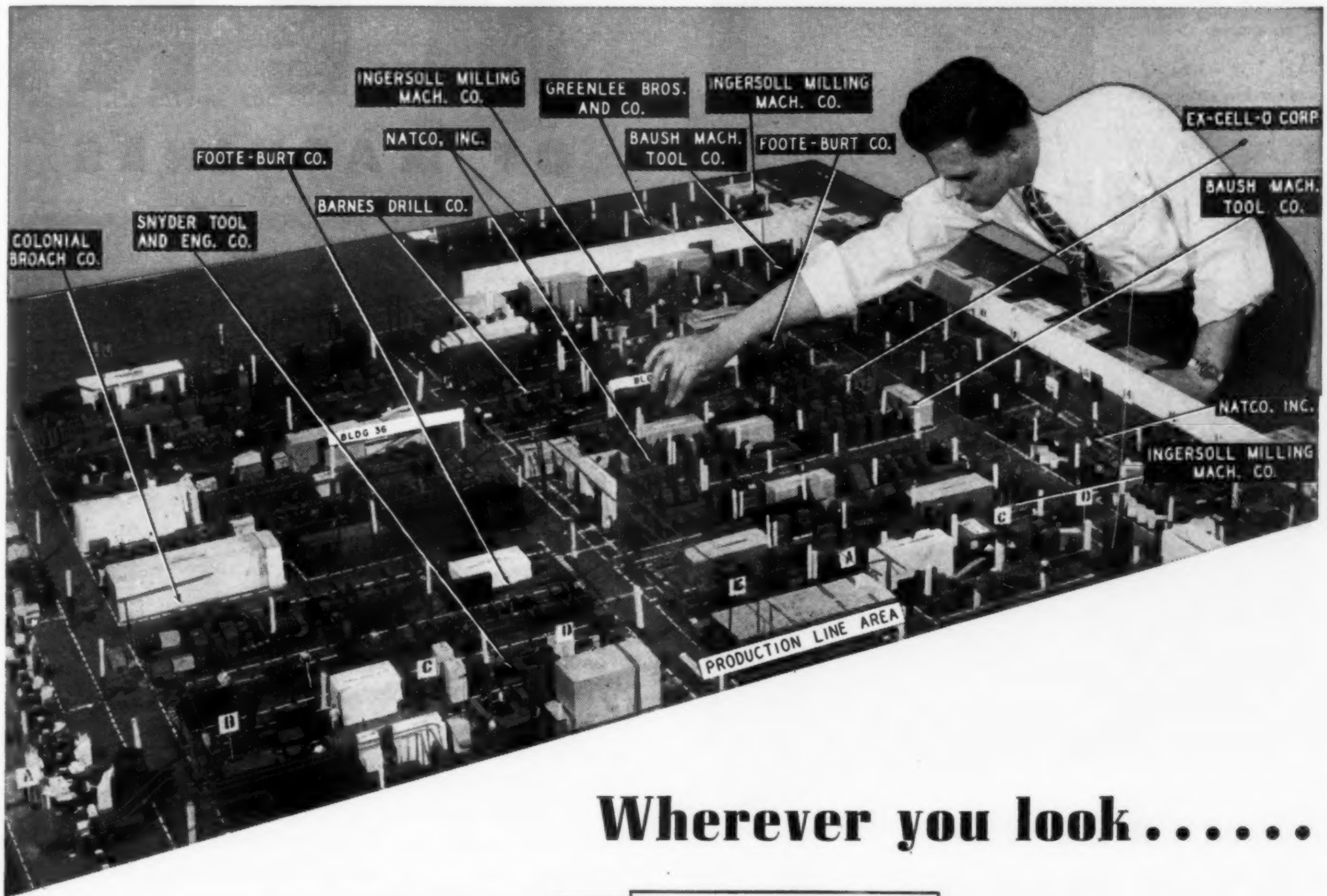
Testing and Inspection

Tinplate Scale: Model 372-S Thermoseal tinplate scale measures the thickness of tin plate by means of a comparison of the weight of steel sheets before and after plating. Scale platform is balanced by a counterweight so that a minimum tare weight is applied to the scale. Scale head is built around two precision calibrated steel springs, and because of its thermostatic control, will automatically adjust to zero over a wide range of temperatures. Capacity is 50 oz. Stainless steel platform measures 20 x 20 in. *John Chatillon & Sons, New York, N. Y.*

Air Gage: For inspection of outside diameter, cloverleaf and taper of cylindrical parts such as automotive piston pins. Part is placed on two steel cylindrical guide rails and pushed through the air ring, which has three open air jets equally spaced 120 degrees apart. Position of the float in the gage column indicates whether or not the pin is within dimensional tolerance. Steel guide rails on each side of air ring can be revolved when sliding surfaces become worn. Standard gages have amplifications of 1000:1, 2000:1, 5000:1 and 10,000:1, depending upon part tolerance; others made to specifications. *Sheffield Corp., Dayton, O.*

Ultrasonic Measuring Unit: Model MS-101 Ultrasonic Metroscope is used for nondestructive ultrasonic measuring and testing of metals and other materials. Makes wall thickness measurements or tests from one side of the material only, as in the forming or construction of propeller blades, tanks, pressure vessels and cylinders, pipes, or any formed or drawn shape. Also detects internal laminar defects in sheet materials and imperfect bonds between clad metals, switch contacts, bearing linings, etc. *J. W. Dice Co., Englewood, N. J.*

In this Modern Automobile Engine Plant



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Wherever you look in this plant you'll see Vickers Hydraulics. This is indicated on the photograph by the lines leading out from the Vickers-equipped machines to the names of their makers. It is an interesting study . . . first for the variety of machines . . . second for the large number of builders who use Vickers Hydraulics. Vickers Hydraulics is often the ideal solution to the myriad problems of feed, control, transfer, synchronization, etc. which must be dealt with in a highly automatized plant such as this. Hydraulic power can be developed wherever most convenient and applied at any other location. Simplification of design, flexibility of control, instantaneous response, rapid operation, ease of providing interlocks and overload protection, low maintenance . . . all are characteristics of Vickers Hydraulic Systems.

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MEN OF MACHINES

J. C. Owen

J. C. Owen was recently appointed chief engineer, instrument products, of the Grand Rapids Div. of Lear Inc., Grand Rapids, Mich. He will direct all engineering activities involving instruments, concentrating on the development of new products and the broadening of the current line of aircraft instruments. A graduate of the United States Naval Academy, Mr. Owen was associated with the Eclipse-Pioneer Div. of Bendix Aviation Corp. for a period of 12 years prior to joining Lear. At Bendix he served as senior engineer in charge of systems engineering activities. Mr. Owen is credited with a number of major contributions in the field of automatic flight.

•

Assistant to the chief engineer since May 1953, **J. P. Cummings** has been named chief engineer of Chrysler Corp. of Canada. Following his initial assignment to experimental work on Dodge trucks when he joined Chrysler in 1930, Mr. Cummings served in various engineering capacities, subsequently becoming resident engineer at the Dodge main plant. During the war years he was assigned to production control engineering on military vehicles built by Dodge. Named chief engineer of the Export Div. in 1945, he served in that capacity until his appointment to the Canadian firm last year.

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Henry J. Gardner has been appointed chief engineer of Airmatic Valve Inc., Cleveland. Previously, Mr. Gardner was associated with the Cleveland Diesel, Chevrolet and Cadillac divisions of General Motors Corp., Electric Auto-Lite Co. and International Business Machines Inc.

•

Raymond Engineering Laboratory, Middletown, Conn., has announced the appointment of **Lincoln Thompson** as a vice president. With a broad background in teaching, research and the development of

talking pictures and electronic devices, Mr. Thompson made the first practical talking book records for the blind and later developed the Sound Scriber disk electronic recorder. He wrote the article, "Dictating Machines," describing the Sound Scriber, which appeared in the January 1951 issue of MACHINE DESIGN.

♦

Philip M. McKenna

Philip M. McKenna, president of Kennametal Inc., Latrobe, Pa., was awarded the Holley Medal by the American Society of Mechanical Engineers at the society's recent Annual Meeting. This award is made for a "great and unique act of genius of an engineering nature that has accomplished a great and timely public benefit." Mr. McKenna was cited for his "research, development and application of cemented carbide compositions which have contributed so much to the art and science of metal cutting." He was also selected to deliver the Towne Lecture at the ASME meeting.

After graduating from George Washington University in 1921, Mr. McKenna did research work toward the development of the best possible tool material. He continued his experiments as research director and vice president of Vanadium Alloys Steel Co., developing a cemented carbide composition superior to previous tool materials. In 1938 he founded the McKenna Metals Co., which later became Kennametal Inc.

♦

Dr. Paul F. Chenea, who came to Purdue University from the University of Michigan to assume the post of professor of engineering mechanics and research professor of materials, was recently named assistant dean of the five engineering schools of the university.

♦

The establishment of a Specialty Control department at the Schenectady, N. Y. works of General Electric Co. and the appointment of **Louis T. Rader** as its general manager have been announced by the company's Switchgear and Control Div. Dr. Rader's staff includes **H. L. Palmer,** manager of engineering.

♦

The appointment of **O. G. Haywood Jr.** as manager of engineering planning has been announced by Sylvania Electric Products Inc., New York. In co-operation with executives of the operating divisions and research laboratories, Dr. Haywood will co-ordinate engineering planning in the fields of lighting, radio,

electronics and television, with particular emphasis on long-range planning.

•

During the annual Medal Day Ceremonies of the Franklin Institute of the State of Pennsylvania, **William Francis Gibbs,** naval architect and marine engineer, was awarded the Franklin Medal, highest honor given by the Institute. Mr. Gibbs received the medal for his efforts in the fields of naval engineering and architecture which ultimately led to the design of the vessel *S. S. United States.*

At the same time, **Dr. Adolph Meyer** was awarded the George R. Henderson Medal for his accomplishments in the gas turbine field, particularly for his part in the development of the first successful gas turbine locomotive.

For his paper, "Compensation of Feedback-Control Systems," **Dr. George Cheney Newton Jr.,** associate director of the Servomechanism Laboratory, Massachusetts Institute of Technology, received the Louis E. Levy Medal awarded by the Franklin Institute.

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The appointment of **James F. Eversole** as vice president in charge of research has been announced by Bakelite Co., a division of Union Carbide and Carbon Corp., New York. Dr. Eversole joined the company in 1929.

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George M. Anderson has been appointed head of the engineering development group at the Edison Laboratory of Thomas A. Edison Inc., West Orange, N. J.

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Five engineering promotions were recently announced by Worthington Corp., Harrison, N. J. **Everett C. Schmactenberg** has been appointed chief engineer of turbomachinery engineering services on centrifugal compressors, high-speed centrifugal boiler feed pumps and marine centrifugal pumps. **C. J. Tullo,** now chief engineer of the Centrifugal Engineering Div., will be responsible for all engineering services on large capacity pumps, vertical turbine pumps, axially split casing pumps, standard end suction centrifugal and refinery pumps. **Warren H. Fraser** was appointed assistant chief engineer of the Centrifugal Engineering Div., as was **William C. Krutzsch Jr.,** who will be responsible for all standard pumps. **Max Reimer** was named chief draftsman of the Harrison Works engineering department and manager of the Engineering Services Div.

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Dr. Adair Morrison has been named head of the research section of the research and engineering department of Sprague Electric Co., North Adams, Mass.

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The American Welding Society has re-elected **Fred L. Plummer** as its president for 1953-1954. Mr. Plummer is director of engineering of the Hammond Iron Works, Warren, Pa.

THE ENGINEER'S

Library

Recent Books

Applied Elasticity. *By Chi-Teh Wang, professor of aeronautical engineering, New York University;* 365 *pages,* 6 *by* 9 *inches, clothbound; published by McGraw-Hill Book Co. Inc., New York; available from* MACHINE DESIGN, $8.00 *postpaid.*

This textbook is intended mainly for design engineers and presumes a knowledge of calculus. Whenever higher mathematics is involved, it is derived where first encountered. Chapters are headed under such titles as analysis of stress and strain, stress-strain relation and elasticity general equations, plane-stress and plane-strain problems, torsion of various shaped bars, finite-difference approximations and relaxation method, energy principles and variational methods, solution by means of complex variables, bending and compression of bars, elastic stability, numerical methods in determination of buckling loads, bending and buckling of thin plates, and theory of thin shells and curved plates.

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Synchros Self-Synchronous Devices and Electrical Servo-Mechanisms. *By Leonard R. Crow, director of research and development, Universal Scientific Co.;* 5½ *by* 8½ *inches,* 232 *pages, clothbound, published by the Scientific Book Publishing Co., Vincennes, Ind.; available from* MACHINE DESIGN; $4.20 *postpaid.*

This book discusses the basic self-synchronous electrical mechanisms and many special synchros and devices that evolved from the fundamental forms. Principles underlying the functional operating theory are given as well as the use and application of synchros and allied self-synchronous electrical mechanisms. Structural features are described in sufficient detail for understanding of the function and application of the many types and forms of self-synchronous devices.

•

ASME Handbook—Design. *Edited by Oscar J. Horger, chief engineer, Timken Roller Bearing Co.;* 420 *pages,* 7½ *by* 10 *inches, clothbound; published by McGraw-Hill Book Co. Inc., New York; available from* MACHINE DESIGN, $10.00 *postpaid.*

Dealing primarily with the design function in metals engineering, this volume is comprised of 48 sections contributed by authorities writing on subjects in which they gained wide recognition. Valuable in-

formation is presented for relating considerations of material selection, design procedure, strength properties, and processing operations for good design. This volume covers essential properties which need to be evaluated by the design engineer in his selection of one material in preference to another.

In Part 1 the overall problem of selection of materials is discussed in general terms. Made up in eight sections, Part 2 covers such items as high-temperature considerations, flame strengthening, plasticity, residual stresses, vibration, fatigue, surface finish, shot peening, cold working, nitriding, joints, wear and impact. Part 3 deals with the problems of corrosion, while a review of present metal testing information is covered in Part 4. The final two parts point out the requirements for mass production and surface finish along with basic information on design theory and practice, elasticity and failure, stresses, and strain gages.

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Temperature Measurement in Engineering. *By H. Dean Baker, professor of mechanical engineering, Columbia University, E. A. Ryder, consulting engineer, United Aircraft Corp., and N. H. Baker, research assistant, Columbia University; published by John Wiley & Sons Inc., New York; available from* MACHINE DESIGN, **$3.75** *postpaid.*

The first of two volumes, this book accurately discusses temperature in terms of engineering measurement. Stress is placed on specific procedures and techniques involved in producing satisfactory temperature-measurement designs for various circumstances. Types of measurement conditions encountered are classified on a physical basis such as interior points in solids, liquids, gases, flame, etc., rather than specific industrial problems or relations to various types of instrumentation. Volume I deals primarily with the thermocouple technique since it is most widely employed to measure internal temperatures of solid bodies.

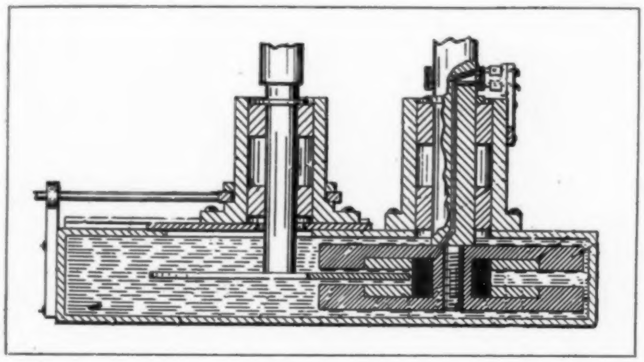
Manufacturers' Publications

Ideas-Techniques-Designs. 306 *pages, 6 by 8½ inches, paperbound; available from Alden Products Co., Dept. HB, 117 N. Main St., Brockton 64, Mass., on company letterhead request.*

This 1954 edition of the Alden handbook is written for engineers and designers of electrical or electronic equipment. The use of standard components is emphasized and design principles of plug-in packages and basic chassis for unitizing equipment in this field are outlined. Methods of indicating and monitoring operation of electronic equipment with tiny indicator lights are described. Electrical wiring connectors and interconnecting systems with color coding are included. This handbook is divided into 11 tabbed sections which contain many illustrations.

NOTEWORTHY
Patents

VARIABLE-SPEED CONTROL of rotating shafts, regardless of direction of rotation, is accomplished with a novel magnetic-fluid clutch design assigned to the Electronic Engineering Co. Covered in patent 2,640,364, the clutch employs a rotating iron disk mounted to slide in and out between two magnetic coils connected to the output shaft and immersed in a solution of oil and iron particles. Excitation of the coils freezes the magnetic particles in solution, creating a magnetic drag to transmit torque from the

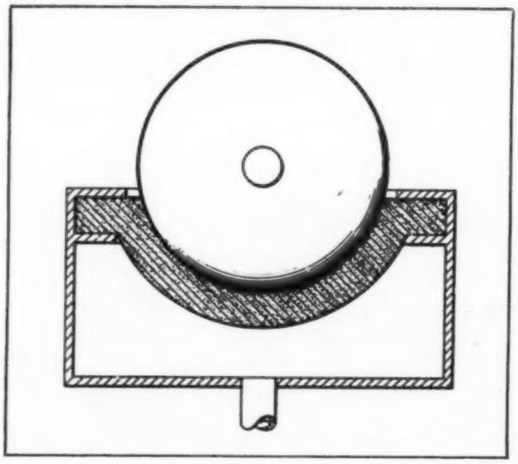


rotating disk to the output shaft. By varying the position of the disk to change the effective radius at which the magnetic forces act, speed of the output shaft can be varied from a minimum to a maximum value, increasing as the disk is moved away from the center of rotation of the coils. Actual power delivered is not affected by disk position and will remain constant at a value directly proportional to the input. A modification of the design for reducing heat losses is also shown in the patent which has been assigned by Gorman R. Nelson.

FLUID-TIGHT SEALING of rotating or reciprocating shafts is accomplished by an oil seal shown in patent 2,646,295 without the usual close tolerance fits required. Designed by John H. Victor, the seal has an outer U-shaped metal casing, press fitted into the housing, and an inner rubber sealing element which engages the shaft and is tensioned by a garter spring. Takeup adjustment to permit greater fit tolerances and compensate for diameter variations in assembly is provided by serrations, a few thousandths of an inch in height, in the outer metal surface. The serrations also act to pilot the seal into the housing bore in a centered position. Necessity for centerless grinding and burnishing operations is eliminated by the design which has been assigned to Victor Mfg.

& Gasket Co. A modification to provide resilient engagement of the outer surface is also described.

Air lubrication by means of porous graphite liners increases load capacity in a hydrostatic bearing design assigned to General Electric Co. by Howard I. Becker. Particularly suited for use with ball-and-socket joints, especially where operating conditions prohibit oil lubrication, the design detailed in patent 2,645,534 utilizes a metal ball rotating in a cup-shaped graphite bearing surface mounted in a sealed pressurized air chamber. Porous openings in the graphite distribute the air pressure evenly over the entire bearing surface, providing a uniform lubricating effect and minimizing the possibility of air shutoff due to overload or clogged air passages. In addition, the natural lubricating qualities of the graphite prevent scoring of the ball surface should contact occur. Several modifications of the basic design, including sleeve and thrust bearing adaptations, are also discussed in the patent.



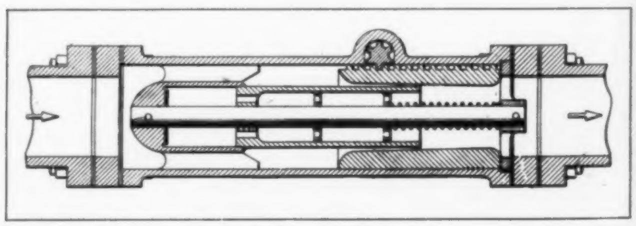
Accurate speed indication for control of dc motors operating on rectified ac voltage can be obtained electrically with a circuit designed by Oscar E. Carlson. Detailed in patent 2,649,572, the circuit incorporates a smoothing filter which produces a steady value of voltage, directly proportional to the back emf of the motor. By calibrating a voltmeter dial in rpm, direct and constant indication of speed can be obtained without the conventional fluctuations associated with rectified current. A modified version of the circuit for a bidirectional control system is also shown in the patent.

Overload torque protection for chain drives, regardless of direction of rotation, is afforded

Noteworthy Patents

by a combination sprocket-clutch design covered in patent 2,643,530. Assigned to Lathrop-Paulson Co., the design can be preadjusted to disconnect at a specified load. In normal operation, torque is transmitted to a tubular drive shaft by a rigidly mounted clutch plate driven by a second matching spring-loaded plate which carries the sprocket wheel and is free to rotate or slide on the shaft. Clutching action is produced by conical stud projections on the driven plate which engage mating depressions on the face of the driving plate. When the load becomes excessive, the driving plate rides over the conical projections, causing the clutch to slip until loads return to normal. Adjustment of the spring pressure, which determines the disconnect load, is accomplished with a collar threaded to the shaft and does not require disassembly. Assignor of the patent is Harry D. Lathrop.

CONSTANT FLOW RATE in fluid lines is maintained automatically, even with pressure variations, by a valve assigned to Ralph N. Brodie Co. under patent 2,647,531 by William F. Berck. Valve operation is based on the Bernoulli principle; a spring-loaded piston mounted in an adjustable converging throat is actuated by fluid velocity to control the flow opening. Piston position is a function of the flow velocity and



is independent of pressure differentials across the valve. Increase in rate of flow causes the piston to move into the nozzle, decreasing the opening and maintaining a constant quantity of fluid at the valve outlet. Decrease in flow rate causes a reverse piston movement. Initial adjustment of flow rate is accomplished by changing the position of the movable throat through a rack and gear mechanism which is operated externally without requiring disassembly.

DROPTIGHT SEALING of antifriction bearings is attained with a clever sheet-metal construction designed to accommodate shaft endplay. Assigned to Saywell Associates under patent 2,642,298, the seal employs overlapping annular sheet metal rings, approximately 0.003-inch thick, to provide a flexible sealing action. The rings are mounted in pairs and O-ring spacers, or other suitable shapes, are utilized to maintain sealing engagement between the overlapping edges. Shaft endplay is absorbed by deflection of the flexible rings without disrupting the sealing action. Assignor of the patent is L. G. Saywell.